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What the Tourist Trailer Means to the Car Manufacturer

By James H. Booth
Buick Motor Co.

AUTOMOBILES used for towing trailers present many overload problems to the car designer. Results of these overloads often do not make their appearance until considerable mileage has been accumulated by the car and trailer. The average car owner is not aware that towing a trailer imposes the same load on all driving members of the car as that of a 20-passenger bus.

The car designer not only realizes these overloads, but also is faced with the problem of providing a suitable hitch that will permit the bumper to function as normal, and the trunk door to open. The author believes that the time is past when it is necessary to destroy the appearance of an automobile by cutting bumpers and substituting ugly pieces of iron to accommodate a trailer.

Closer cooperation between the car designer and trailer designer is needed to overcome the difficulties when a trailer is involved. Height of hitch, point of attachment, addition of extra spring leaves, tires, tire pressures, rear axle and gears, transmission, shock absorbers, brakes, cooling, gasoline consumption, electrical system, and all the units in the chain of parts comprising a finished automobile are affected by towing trailers.

I HAVE been requested to present the problems of the automobile manufacturer resulting from the increased use of automobiles for towing house trailers. If it had been suggested that I write a list of nightmares for the car designer and classify them in their order of intensity, my task would be simplified. Every additional 100 lb. of weight is a nightmare for the car designer and, if 2500 lb. or more of trailer are suddenly added to his problems, the designer may be expected to experience unpleasant dreams.

It would not be a tremendous undertaking to design a car for trailer towing, but there is not sufficient volume to warrant a special design, and we are all aware that automobile owners are continually using the standard design cars for towing trailers.

If any of us should contact almost any car dealer and ask him if the automobile that he is selling would tow a trailer, we would receive an affirmative reply. Without sufficient knowledge of the overloads involved, most car dealers blindlv reason that, if competitive cars are towing trailers, there is no reason why the particular make of car that they are selling cannot perform this type of service. These dealers are correct in their attitude to some extent and should be admired for their faith in automotive products. Automobiles will tow trailers, but the number of miles that they will tow them without costly repairs is another matter. Let us assume that, after contacting the car dealer regarding trailers, we again approach him with a proposition to buy his car and convert it into a 20-passenger bus. This same dealer would inform us immediately that we would accept delivery of the car without any responsibility on the part of the car manufacturer if the car were loaded beyond its designed capacity. However, except for static load on the chassis, these two illustrated cases are identical. The load on the engine, clutch, transmission, driveshaft, differential gears, and axle shafts, is the same for 3000 lb., whether the load is 20 passengers of 150 lb. each or 2500 lb. with two car passengers and 200 lb. of baggage.

The car designer realizes these overloads and the manner in which they will make their appearance as the cars accumulate mileage. The designer also knows that he cannot remedy all of his headaches by the aspirin method of substituting a larger unit or increasing the section of one badly overloaded member. Some units just cannot be helped and will have to suffer until the trailer industry absorbs a sufficient percentage of automobile production to warrant special equipment on cars intended for this purpose. We cannot condemn any automobile manufacturer for refusing responsibility when the car he so carefully designed to function as an automobile is converted into the equivalent of a truck. At the present time we

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., May 7, 1937.]

find automobile designers testing cars with trailers, and some engineering departments are making all possible efforts to serve the car owner and the trailer manufacturer by providing designs of trailer hitches and recommendations that will prolong the life of automobiles in this service. There are many viewpoints to these problems, and closer cooperation between the car designer and trailer designer is needed to overcome the difficulties where the trailer is involved in the problem. In all engineering of this type there is compromise and, as present volume does not permit the design of a special car, it will be necessary for the trailer manufacturer and the car manufacturer to make minor alterations in design for the ultimate benefit of both.

Before the public desire for rapid acceleration, high speed, a soft ride, and faster deceleration influenced design greatly, the addition of a trailer to the passenger car presented only a minor problem. Cars designed a few years ago were much heavier, compression was lower, motor revolutions were down, and spring rates were up. The car designer used safety factors in proportion to the car weight, and this weight was high enough so that the addition of a trailer did not show as such a large percentage of the whole. The present design of cars presents us with an automobile designed to function in the most efficient manner, and no excess weight is tolerated.

Makeshift Hitches Dangerous

The first problem of the car designer is to provide a suitable hitch. I am not aware of any special passenger-car frame designed for a trailer hitch, and I do not foresee sufficient volume to warrant the burden on the total automobile production necessary to provide such a frame. The car designer will do well to visit a trailer camp before starting his hitch design, where he will find some good hitches and many that should be ruled from the highway. I have seen hitches fastened to the middle section of bumper bars, some fastened to spare-tire carriers, and others just wrapped around bumper brackets in every conceivable manner. These hitches are dangerous. Bumper bars are designed for collision and will not endure the fatigue of continued reversal of load caused by a trailer hitch.

Another type of hitch that we commonly see is the one constructed of flat bar stock or angle iron, fastened to the bumper brackets, and with the bumper cut away for the accommodation of the trailer drawbar. This hitch is better than the one that uses the bumper bar, but it is far from a desirable method. I believe that the time is past when it is necessary to destroy the appearance of an automobile by cutting bumpers and substituting ugly pieces of iron to accommodate a trailer. In addition to appearance this construction has other disadvantages. The cross-bar is fastened only to the side-rails of the frame and, although the construction is fairly rigid with respect to accelerating and braking loads, it is usually weak on vertical loads caused by the inertia of the trailer negotiating bumps and by the drawbar load.

Use of Car Without Trailer

Another disadvantage of this type of construction is found in the use of the car without the trailer. If the car operator backs the car into a pole or other massive obstruction, this hitch is quite rigid in horizontal loading and will not deflect as will the original bumper bar. The load is transferred to the frame side-rails, and damage of a costly nature is usually the result.

A design of hitch that we often see is the type where a section of channel iron is fastened to the rear cross-members at the center of the frame. This hitch is suitable for braking and accelerating providing the cross-members are of rigid construction, and also for vertical loads caused by inertia of the trailer, but usually the cross-members of the frame fatigue under constant reversal of concentrated loading at the point of attachment due to unusual deflection, and often break.

Another type of hitch that attaches to the side-rails for accelerating and braking loads, and also fastens to the rear cross-members of the frame for vertical loads, seems to have the most advantages.

Any hitch should be designed to allow normal operation of the bumper and to permit opening and closing of the rear deck or trunk door. The point of attachment of the trailer should be ahead of the bumper to eliminate danger of locking with other cars when the car is used without the trailer. Ground clearance should be considered, and the point of hitch should have as much ground clearance as the lowest point of any sprung member.

Hitch Height Important

The height of hitch is another problem facing the car manufacturer in designing a trailer hitch. At the present time trailer manufacturers have drawbar connections at various heights, and this problem is now before a committee appointed by the Society. If it agrees on a fixed height of drawbar attachment. it will help but not entirely remedy this problem. When we consider the problems formerly discussed in designing a trailer hitch, we immediately realize that each car manufacturer will have his own problems of locating the point of attachment to permit the bumper to function and the trunk door to open.

If an ingenious car designer finally overcomes all the obstacles of sheet metal and bumper and locates the point of attachment at the predetermined height, he is far from finished with this problem. All trailers do not impose the same vertical load on the hitch, and most car rear springs vary in rate among models and makes. The wheelbase affects the load transfer of front to rear in relation to the dimension from the center of rear axle to the point of attachment. For example, a drawbar load of 300 lb. imposed by a trailer on a car with a dimension A from the centerline of the rear axle to the hitch will cause an additional rear-axle load of 400 lb., and lighten the front by 100 lb. Another car with a dimension of $A + 6$ in. will have an entirely different load transfer.

In all cases where trailers cause a load on the hitch they not only impose the drawbar load, but add the load transfer from the front to the rear axle, springs, and tires.

Another factor in this problem is the owner who installs added leaves to his rear springs, or helper springs, and the owner who does not and will not endure the increased harshness of ride caused by the added leaves when the car is used without the trailer.

Solution Suggested

The only solution that I can suggest is to determine a desirable height of hitch and to provide a trailer drawbar fastened to the trailer frame in a manner to permit a limited range of adjustment. The trailer manufacturer should view this problem as the car manufacturer regards his adjustable seat relative to steering-wheel position. The car designer realizes that all persons are not similarly proportioned and that a front seat must have a sufficient range of adjustment to accommodate the massive and slim types of owners relative to their position behind the steering wheel. Cars of various makes have as many varying proportionate parts as does the human anatomy, and a trailer drawbar with an adjustment range would aid the car designer greatly on this problem.

In our discussion of hitches we have considered the transfer of load from front to rear and, after the hitch has been designed, the car designer's next considerations are the rear tires

and rear springs. The rear tires were designed for the usual passenger load and now are overloaded greatly. In most cases they were overloaded slightly with a normal passenger load as most car designers realize that a car is usually used only a minor part of its life with its normal passenger capacity. A survey of tire sizes shows that, with most cars, tires are overloaded 5 to 10 per cent with a normal passenger load.

Added Tire Loads

A trailer, however, imposes a constant static load and a variable inertia load. It seems that we can discard inertia load, except in extreme cases, and consider tires from static load only. Let us consult the tables of the Tire and Rim Association and check these overloads on a popular-sized tire. A 6.50-16 four-ply tire shows a maximum load of 1050 lb. at 28 lb. per sq. in. inflation. If a car has 1102 lb. of load per rear wheel with a normal passenger load we have an overload of 5 per cent. Add a trailer imposing another 200 lb. per wheel, and we have a load of 1302 lb. per tire, or an overload of nearly 25 per cent. If we consult our tables again in search for the proper tire to be substituted, we find that a 6.50-16 six-ply tire is rated for 1215 lb. at 36 lb. per sq. in. inflation, and this tire would be a possibility but, for this constant load, a 7.00-16 should be recommended. We must remember, however, that we not only added 400 lb. to the rear tires but, due to the position of the hitch, we removed 100 lb. from the front tires. We consequently reduced the slip angle and increased the cornering power of the front tires, and greatly reduced the cornering power of the rear tires at normal inflations. We can expect the car handling with this combination to be very poor, and tests have shown that our expectations are realized.

By further testing with tire pressures to restore car handling, equalizing the cornering ability of front and rear tires, we have concluded that front tires at normal production inflation and an increase in rear tire pressure up to 20 lb. per sq. in. above normal will restore the handling ability of the car with the trailer attached. It is apparent that any four-ply tire immediately is discarded from consideration and, in this particular case, a 7.00-16 six-ply tire should be substituted for safety and carrying capacity. Some of the tire manufacturers have tested cars with trailers attached and recommend six-ply tires in all cases, and oversize tires in a few cases, for best car-handling ability.

Springs are the next consideration and, although we may add additional leaves to the present rear springs to carry the increased load, this procedure has the disadvantages of increased harshness when the car is used without the trailer attached and provides an unbalanced car for roadability. This installation is costly to the car owner and, once installed, cannot be removed without inconvenience and expense. Bumper springs have been tried and, in most cases, are unsatisfactory. These bumper springs will carry the additional load with the trailer attached but, when the car is operated without the trailer, most of these springs do not follow rebound and result in a choppy ride.

Removable Helper Spring Needed

There is need for a design of helper spring that may be attached and disconnected with minor expense and inconvenience. At the present time the automobile manufacturer has no alternative other than to recommend adding spring leaves to carry the load, and the trailer owner will have to bear the penalty of increased harshness when he uses the car without the trailer. The shock absorbers do not appear to suffer in proportion to the other chassis units.

The rear axle and gears are the next problem, and here is where the car designer is forced to abandon his aspirin type

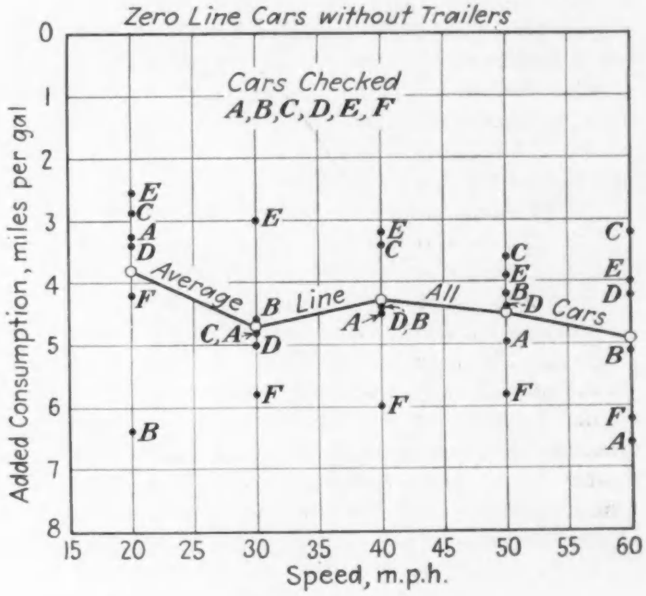


Fig. 1 – Fuel Economy – Six Cars with Trailers

of construction. Axles, bearings, gears, driveshaft, transmission, clutch, and engine cannot be helped as easily as can the units previously mentioned. They all will suffer in direct proportion to the load requirements. Very little can be done for the rear axle or any of these units unless replacements are made. Not only is the load a factor, but the fatigue of some units is increased tremendously. Transmissions used in passenger cars often are used in taxicab service, and the records show that the added number of gear shifts fatigue the gears and shorten transmission life. Cars used with trailers require much more gear shifting than the average car receives as, in traveling hilly or mountainous country, the torque requirements from the motor are increased.

Trailers Hard on Transmissions

I have followed cars with trailers attached over mountainous country and, in many cases, I have observed the car running up hill at 40 m.p.h. in second gear, and descending the following grade in second gear at approximately the same speed to save the use of brakes. This use rapidly fatigues the transmission by imposing heavy reversed overloads on the gear teeth.

We may reasonably expect less transmission life from a car towing a trailer than we receive from passenger cars in taxicab service due to the overload combined with the increased number of gear shifts.

As we accumulate more test miles with trailers attached, we shall discover the weakest links in these units by repetition of troubles. In one car it may be a clutch and, in another, an axle bearing, but we may be sure that the points most affected by overloads will demonstrate their weaknesses. The car designer may profit in testing with trailers by watching these troubles and by building a slightly larger margin of safety into the most troublesome units.

Brakes are not the large problem that they were a few years ago as most of the heavier trailers have a braking system. This provision is a relief to the car manufacturer and a decided help to the trailer owner for safety and freedom from brake trouble. I do not expect much complaint of brakes from car owners where the trailer is equipped with enough braking area.

The cooling system has received very little attention, and what present cooling systems will show when cars with trailers

travel in high altitudes in hot weather I do not know. I can remember a trip a few years ago when a car I was using to tow a trailer boiled all the way across the State of Arizona. At that time we expected a penalty for pulling a trailer, and gave the matter little thought. Car owners of today will not tolerate this penalty, and we should investigate our cooling under these extreme conditions.

The gasoline-consumption problem has been surveyed, and the penalty at various speeds is shown in Fig. 1. The car manufacturer should inform his dealers of this increase so that car customers will know the mileage to be expected.

Electrical systems suffer in direct proportion to the added electrical units of the trailer. Trailers often carry many additional lights, electric brakes, and radios, overloading the generator capacity of the car. Trailer manufacturers may find it necessary to install a separate electrical system driven by the trailer wheels. The automobile manufacturer cannot burden the average car owner with additional expense for a generator of sufficient capacity to supply a trailer.

A previous paper on trailers given at the 1937 Annual Meeting of the Society in Detroit by Philip H. Smith[1] humorously stated as near as I can remember: "Trailers have been going around and around and have not come out any place yet."

In conclusion I should like to state that I heartily disagree with the latter part of this statement. It is true that the trailer has been going around and around, but it did come out – and landed right in the car designer's lap!

[1] See S.A.E. Transactions, February, 1937, pp. 45-47 and 56; "Where is the Trailer Going?" by Philip H. Smith.

European Construction Reviewed in Discussion of Budd Paper

– Maurice Platt
Technical Editor, "The Motor," England

Edward G. Budd's paper, "The Aircraft Trend in Body Structural Design," was published in full on pp. 65 and 66, of the February, 1937, issue of the Journal, *following presentation at the Annual Meeting of the Society, Detroit, Mich., Jan. 13, 1937. This discussion arrived too late for publication with the paper.*

IN his paper Mr. Budd remarks that integral construction of body and chassis has been accorded considerable attention in Europe. There are some very early examples of touring bodies built as one structure with the chassis frame such as the Lancia and the Trojan utility car.

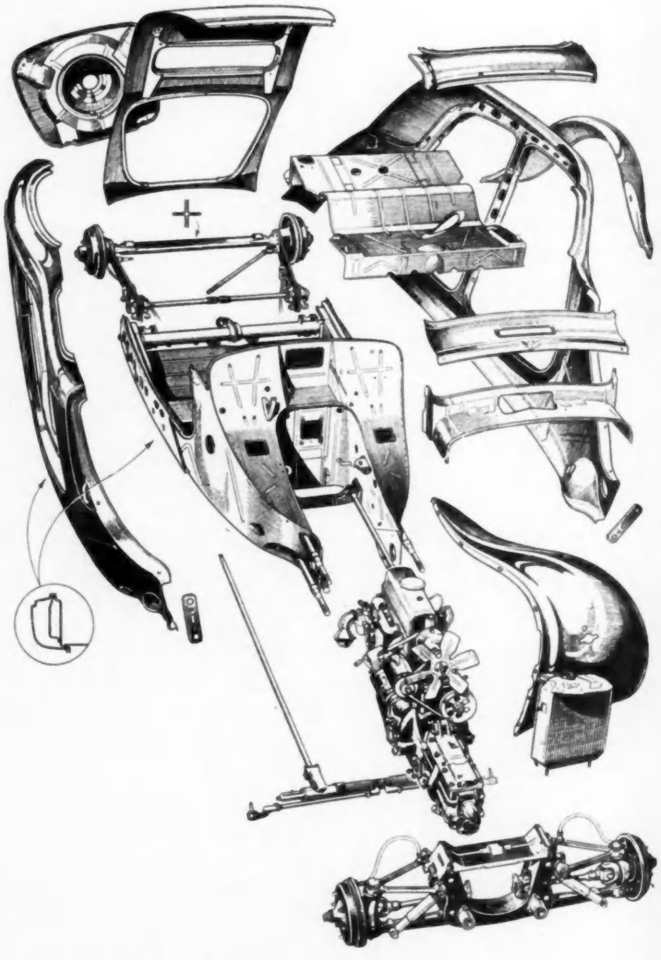
By far the boldest development, however, was represented by the front-drive series of cars initiated by M. André Citroen nearly three years ago. At the Citroen works at Slough, Buckinghamshire, I have seen these cars put together by welding processes from steel pressings which are largely shipped from Paris. These components are shown in Fig. *A*, which is self-explanatory.

Apparently very little trouble is experienced with accidental damage although the completed structure is cellulosed, trimmed, upholstered, glazed, and fitted with instruments before any of the mechanical components is added. One of these components is a unit which comprises the engine, gearbox, final drive, and differential. Another consists of the front suspension, and the third is a light rear axle which trails from radius arms mounted on a transverse tube at the rear end. Torsion bars are employed all round, together with hydraulic shock absorbers of the telescopic type. The completed car is finally tested, inspected, and polished.

The 15-hp. model which we tested on the road not long ago has an unloaded weight of 23¼ cwt., a wheelbase of 10 ft., 1½ in. and sells at £315 in England; these figures refer to the saloon.

It will be noticed that the front-drive principle, by concentrating all the mechanism at one end of the car, is particularly suitable for this type of construction. A rear-drive, rear-engined job obviously would do equally well.

The only other important European example of a car without a separate chassis is the Opel, built in very large quantities by the General Motors plant in Germany. It is a rear-drive model with the engine at the front, and the saloon is a two-door job. The selling price is extremely low. The engine is rated at 11.3 hp., the wheelbase is 7 ft. 8 in., and the weight is 15 cwt.



Courtesy of The Motor, England

Fig. *A* – The Principal Components of the Citroen Front-Drive Car in Which the Body Structure, Welded from Pressed-Steel Parts, Takes the Place of a Chassis – This Model Has Been in Production for Nearly Three Years

High Output in Aircraft Engines

By R. N. Du Bois and Val Cronstedt
Aviation Mfg. Corp., Lycoming Division

RAPID development and widespread use of aircraft fuels of high knock rating make advisable a study of the best means of utilizing these fuels for high specific power output.

The response of an hypothetical aircraft-engine cylinder to changes in supercharger compression ratio (boost) and cylinder compression ratio is developed theoretically and outlined by means of charts. The explosion pressures are calculated and the resulting information used to establish a criterion of cylinder performance.

Results of some cylinder calibrations on high-octane fuels show the calculated relations to be fairly well established.

The effects of (1) high mean effective pressures and (2) high crankshaft speeds on the detail design of the engine are analyzed. The conclusion is reached that, in order to obtain a balanced design of minimum weight for extremely high specific output, it is advisable to use both speed and high mean effective pressures.

THE use of aviation fuels of 100 octane number is becoming fairly widespread. Capt. F. D. Klein of the Materiel Division, U. S. Army Air Corps, has stated that the Air Corps will purchase approximately 2,841,000 gal. of this fuel during the year ending July 1, 1937.[1] The available supplies of technical iso-octane are estimated at 155,000,000 gal. per year, and it is possible to produce 340,000,000 gal. of technical isopropyl ether yearly, according to Messrs. Buc and Aldrin of the Standard Oil Development Co.[2] The total estimated consumption of all aviation gasoline in the United States for 1936 is under 100,000,000 gal., so that it is apparent that there is ample potential supply of these very high octane blending agents. Both materials are susceptible to the addition of tetraethyl lead to increase their knock rating to values above 100 octane. It appears that fuels of still higher antiknock value will thus be available when demand arises.

It is, of course, evident that most of the 3,000,000 gal. of high-knock-rating fuel to be consumed in 1937 will be used by engines whose designs were developed on much poorer fuels and which have been increased gradually in output by detail improvement to make use of the increasing quality of the fuels.

It will be attempted in this paper to discuss the possibilities of the high-octane fuels from the standpoint of the designer who might be permitted to start from scratch and proceed to develop his basic design around them.

Theoretical Considerations

Fig. 1 shows the well-known curve of air-cycle efficiency vs. volumetric compression ratio, using the exponent $\gamma = 1.396$. It has been stated by Pye[3] and confirmed by others, that the thermal efficiencies obtained in practice are about two-thirds of these values. Such a curve has been drawn in Fig. 1. The curve of power output vs. compression ratio will, in the absence of detonation, follow this thermal efficiency curve if running unboosted, or "normally aspirated" as we painfully put it. Changes in compression ratio of supercharged engines affect the volumetric efficiency by changing the volume of residual gases which are compressed to the supercharge pressure during the intake stroke.

Referring to Fig. 2, the increase in volumetric efficiency with

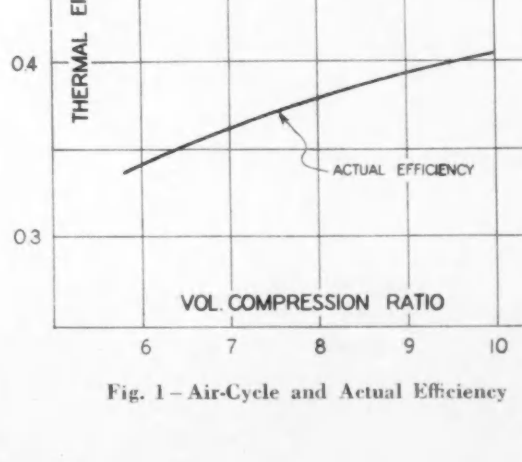


Fig. 1 – Air-Cycle and Actual Efficiency

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 15, 1937.]

[1] See S.A.E. TRANSACTIONS, August, 1936, pp. 304-312; "Future Possibilities of 100-Octane Aircraft-Engine Fuel," by Capt. F. D. Klein.

[2] See S.A.E. TRANSACTIONS, September, 1936, pp. 333-340 and 357; "A New High-Octane Blending Agent," by H. E. Buc and Major Edwin E. Aldrin.

[3] See "The Internal-Combustion Engine," by D. R. Pye, Oxford University Press, 1931, pp. 19 and 20.

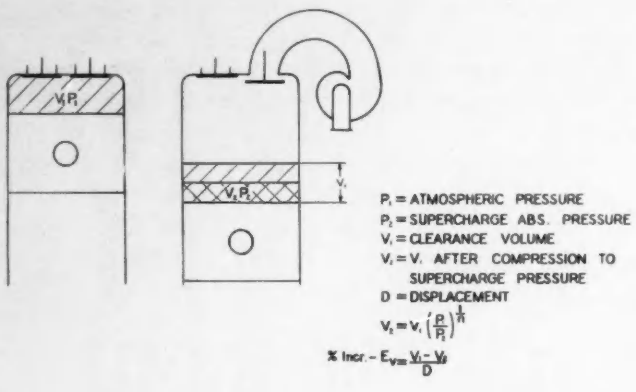


Fig. 2 – Compression of Residual Gases

increasing supercharge pressure may be demonstrated as follows, where:

P_1 = atmospheric pressure.
P_2 = supercharge absolute pressure.
V_1 = clearance volume.
V_2 = V_1 after compression to supercharge pressure.
D = displacement.
n = exponent of polytropic compression.

$$V_2 = V_1\left(\frac{P_1}{P_2}\right)^{\frac{1}{n}}$$

$$\text{Per cent increase} \quad E_V = \frac{V_1 - V_2}{D}$$

It is obvious that increasing the compression ratio at a given boost pressure decreases the clearance volume V_1 and hence reduces the value of the term $V_1 - V_2$. Fig. 3 shows the relative volumetric efficiencies for various supercharge absolute pressures and cylinder compression ratios, assuming adiabatic compression of the residual gases to the supercharge pressure.

As a basis for our performance calculations, a cylinder developing 100 i.hp. at 60 deg. fahr. and standard atmospheric pressure at the air intake with a 6:1 compression ratio, has been chosen. The atmospheric line in Fig. 4 shows the power increase due to increased thermal efficiency alone. The other curves show the increase in indicated power to be expected at various boost pressures after allowing for the changes in volumetric efficiency just described, at constant manifold temperature. However, the supercharger has been found to heat the air passing through it to the equivalent of polytropic compression with "n" = 1.6. The resulting manifold temperatures are shown in Fig. 5, assuming the air entering the supercharger to be at 60 deg. fahr. and neglecting heat gained or lost by the air or charge between the supercharger outlet and cylinder intake port.

The effect of air temperature on power is known to follow the "inverse-square-root" relation in which the indicated power varies inversely as the square root of the absolute temperature. Using this rule and the temperatures of Fig. 5, we may correct Fig. 4 and obtain the actual indicated powers to be expected with the compression ratios and manifold pressures shown (see Fig. 6).

We have been dealing thus far in indicated output but, since it is shaft or brake power that turns the propeller, the friction and supercharger drive losses must now be considered. The power to drive the supercharger is dependent upon the mass flow of air through it, the boost required, and the efficiency of the unit as a compressor. The engine friction losses, as distinct from the supercharger drive power, may be considered as unaffected by changes in compression ratio, although it is probable that in most engines there is a slight increase in friction power with higher compression. When measuring friction power on a single-cylinder engine, it is found that, with valve timing as used on high-performance cylinders, increasing the supercharge pressure decreases the friction mean effective pressure by about 0.45 lb. per sq. in. for every pound increase in supercharge pressure, due to the force exerted on the piston by the supercharge pressure during the suction stroke. This constitutes an appreciable reduction which may be applied to the computed supercharger drive power.

In computing the total engine friction per cylinder plotted in Fig. 7, it has been assumed that the engine at 6:1 compression ratio will have friction losses, unboosted, amounting to 15 per cent of the indicated power. Also, it has been assumed that the air consumption for purposes of computing the super-

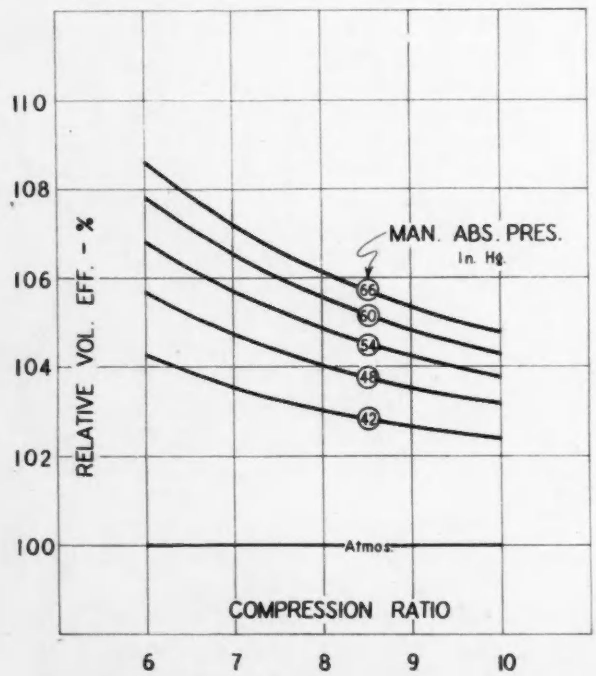


Fig. 3 – Relative Volumetric Efficiencies

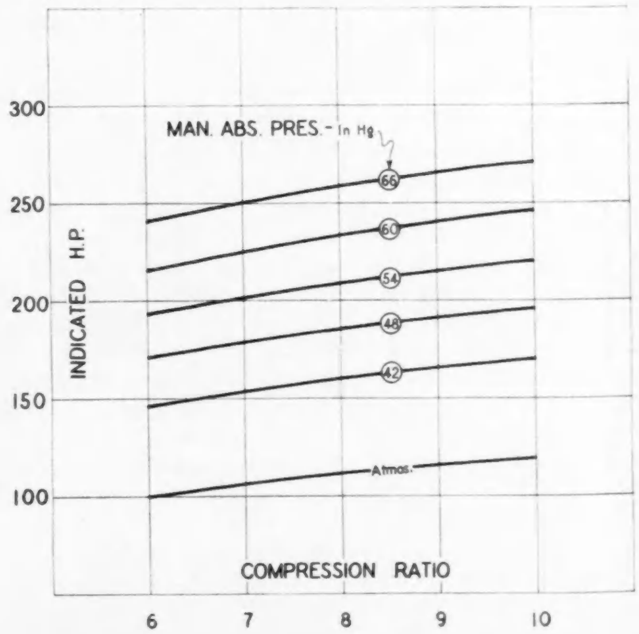


Fig. 4 – Indicated Horsepower Vs. Boost and Compression Ratio at 60 Deg. Fahr. Air Temperature

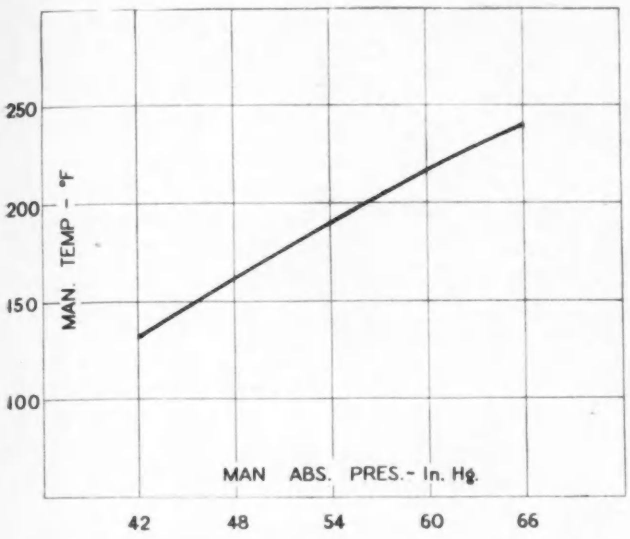


Fig. 5 – Supercharger Outlet Temperature Vs. Boost at 60 Deg. Fahr. Inlet Temperature

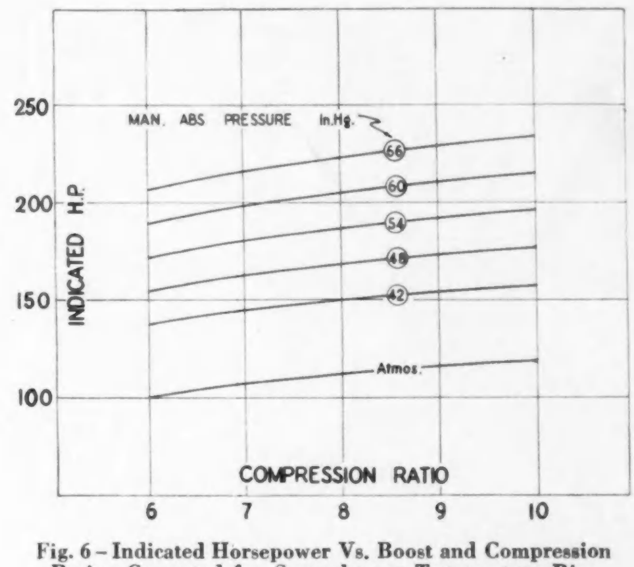


Fig. 6 – Indicated Horsepower Vs. Boost and Compression Ratio – Corrected for Supercharger Temperature Rise

charger drive power is proportional to the indicated power, and that the efficiency of the supercharger as a compressor is 65 per cent. It is found that the overall mechanical efficiency of the engine-supercharger combination remains very close to 85 per cent over the range of pressures and compression ratios which have been used in the computations.

Fig. 8, showing the brake power corresponding to the indicated power of Fig. 6, has been drawn using the total engine friction of Fig. 7.

It should be emphasized here that each of the manifold pressures shown is to be obtained by application of a supercharger designed to produce that particular pressure at the rated speed of the engine.

The effect of supercharging and compression ratio on cylinder pressure is shown in Fig. 9, in which explosion pressures, calculated by Prof. A. J. Meyer of the University of Kentucky, have been plotted.[4] The rapid increase in these pressures with boost at the higher compression ratios is noteworthy. The explosion pressures may be added to the information of Fig. 8 to obtain Fig. 10. This figure looks like a Mollier diagram, but shows the brake power output and explosion pressure of our hypothetical cylinder for any boost or compression ratio under consideration.

Practical Applications

It is evident from an examination of these curves that a certain specific power output may be obtained in a number of different ways. For instance, a specific output of 132 b.hp. per cylinder at 48 in. absolute pressure and 6:1 compression ratio will also be obtained at 37 in. absolute pressure and 10:1 compression ratio provided the design will permit an explosion-pressure increase from 1040 to 1600 lb. per sq. in. with the fuel for which the cylinder is intended to be used.

It is believed that the limiting maximum explosion pressure of a given cylinder design on a specified fuel constitutes a very useful gage of its excellence. It determines the extent to which the compression ratio may be increased when striving for low fuel consumption for long-range aircraft and also indicates the amount of supercharge permissible when the maximum power output is required and increased fuel consumption is allowed. It seems desirable, therefore, as an early step in the development of a cylinder, to establish its limiting maximum explosion pressure on the specified fuel by means of single-cylinder calibration. This calibration may be done at any convenient compression ratio and preferably at intake-air temperatures which are varied with supercharge pressure as shown in Fig. 5. The explosion pressure for the test conditions may be obtained either by measurement or by use of the chart, Fig. 9.

The following results of some calibrations of high-output cylinders may be presented. These tests were made at two compression ratios and with aviation fuels ranging from 87 octane by the Motor Method to iso-octane plus 4 cc. of tetraethyl lead. A uniform intake-air temperature of 200-210 deg. fahr. was used.

For measurement of explosion pressures, the N.A.C.A. maximum-pressure indicator was used. This indicator is a modified "Farnboro" type of balanced-pressure disc-valve, uncooled. Fig. 11 shows the set-up used with this indicator. Fig. 12 shows the valve and its accessories on a cylinder for calibration. In using the indicator, the controlled pressure is applied to the valve and allowed to drop until a point is reached where the neon bulb just commences to flash. This pressure is recorded as the maximum explosion pressure for the given

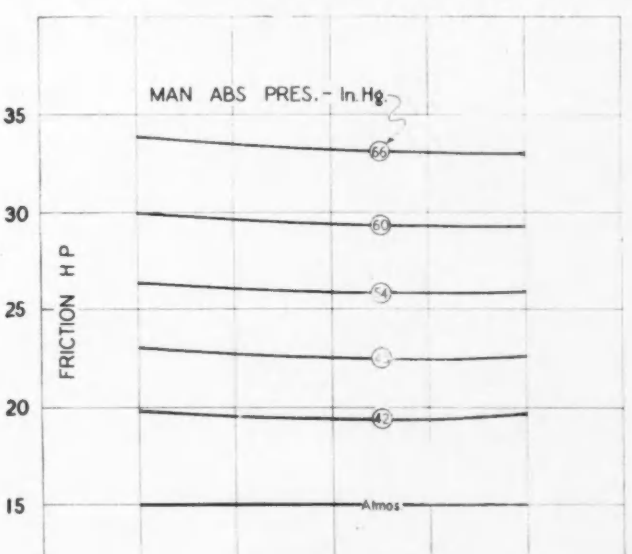


Fig. 7 – Friction Horsepower (Engine Friction Horsepower Plus Supercharger Friction Horsepower)

[4] These pressures have been computed for a chemically correct mixture of octane and air. The corresponding pressures for gasoline containing no octane may be slightly lower.

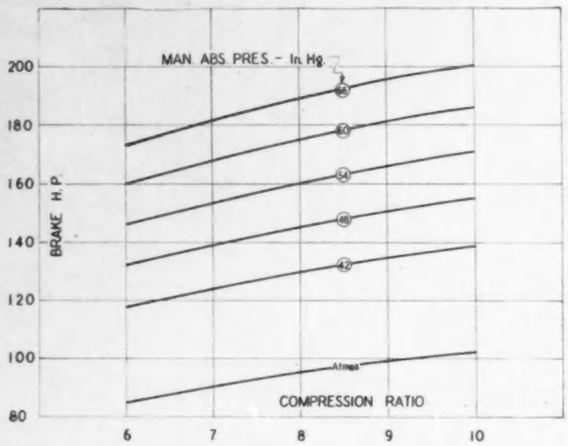


Fig. 8 – Brake Horsepower

conditions. The pressure back of the disc is then allowed to fall until a point is reached where the bulb just commences to flash regularly. This is the minimum explosion pressure. The difference between these two pressures represents the cycle-to-cycle variation in explosion pressure. The compression pressures are obtained during friction runs by cutting the fuel and motoring the engine under the same conditions.

Fig. 13 shows a cylinder calibration with technical iso-octane at 6:1 compression ratio. The cyclic variation in explosion pressure is of the order of 400 lb. per sq. in. and brackets the calculated values as shown. A rough rule (at least for the range of 200-300 lb. per sq. in. i.m.e.p.) for the minimum explosion pressure is 3 times the i.m.e.p.; for the maximum explosion pressure 4.5 to 5 times the i.m.e.p. The indicator was not used at higher compression ratios and boosts due to mechanical difficulties. The mean of the measured explosion pressures decreases with respect to the calculated values due to the reduced spark advance at higher manifold pressure. The great variation in explosion pressure from cycle to cycle is interesting. These variations seem to be characteristic of Otto-cycle combustion. When taking cards with the Farnboro indicator, the scattering of points at the pressure peak caused the indicator to be suspected, until cards at equal pressures and speeds on Diesel cycles showed perfectly uniform lines over the entire cycle.

[5] See N.A.C.A. Technical Report No. 202, 1924; "The Sparking Voltage of Spark-Plugs," by F. B. Silsbee.

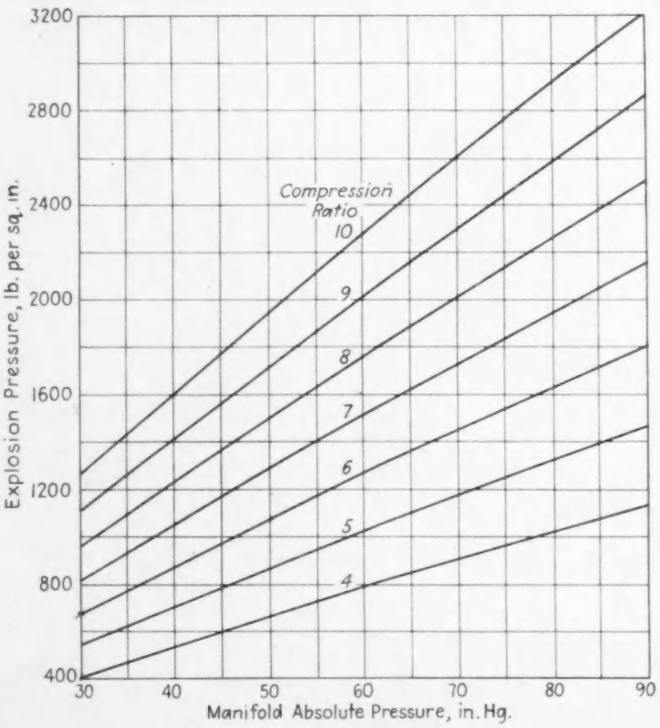


Fig. 9 – Calculated Explosion Pressures

A possible explanation of this cyclic variation in explosion pressures may be the effect of changes in the spark timing. F. B. Silsbee[5] has shown that sparking voltage varies directly with charge density at the time of ignition. This relation has been confirmed in our own laboratory under the higher temperatures and pressures with which we are concerned. Variations in sparking voltage with electrode wire of low electronic emission have been ascribed to the fact that the spark cannot occur until the gap has been ionized and, therefore, must wait

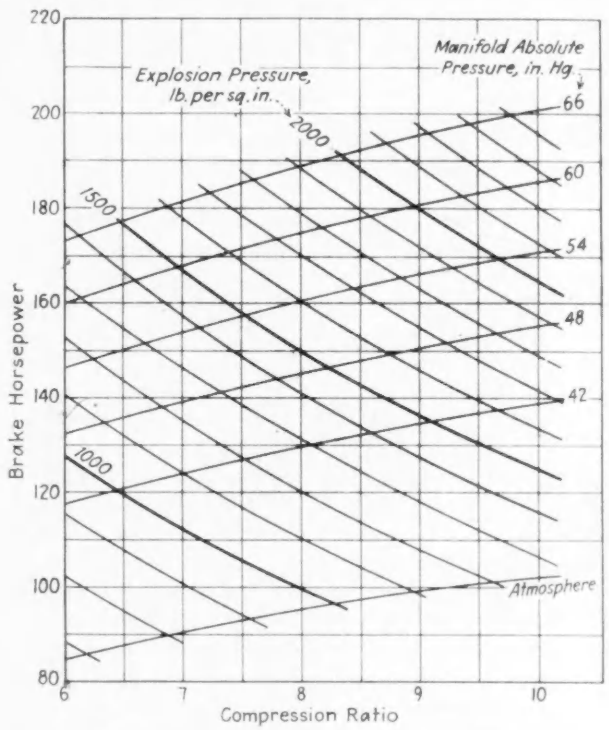


Fig. 10 – Calculated Explosion Pressures Vs. Boost, Brake Horsepower, and Compression Ratio

until an ion "drifts" into the gap. The resulting differences in sparking voltages are due to the gap becoming ionized at different points on the compression line. Certain manufacturers of spark-plugs have kept these variations in sparking voltage very small by the use of electrodes having high electronic emission, and which, therefore, tend to keep the gap ionized at all times. At the engine speed used in these tests, a spark delay of 0.001 sec. would cause a retardation of about 19 deg. which would, of course, greatly reduce the explosion pressure.

In calibrating a cylinder for maximum output on a given fuel, the indicated mean effective pressure is plotted vs. intake manifold absolute pressure at constant speed for several compression ratios. The curve of output vs. boost is a straight line until the limiting output is reached, at which point the curve bends downward and detonation usually becomes audible. Such a calibration is shown in Fig. 14, using 87-octane (Motor Method) aviation fuel, iso-octane, and iso-octane plus 3 cc. of tetraethyl lead. The compression ratio was 6:1. The calculated explosion pressures have been indicated at the limiting output with the first two fuels, and at the maximum manifold pressure used with the iso-octane plus lead. The cylinder was operating without distress with this fuel at 84 in. hg. absolute, manifold pressure, which was the maximum obtainable with

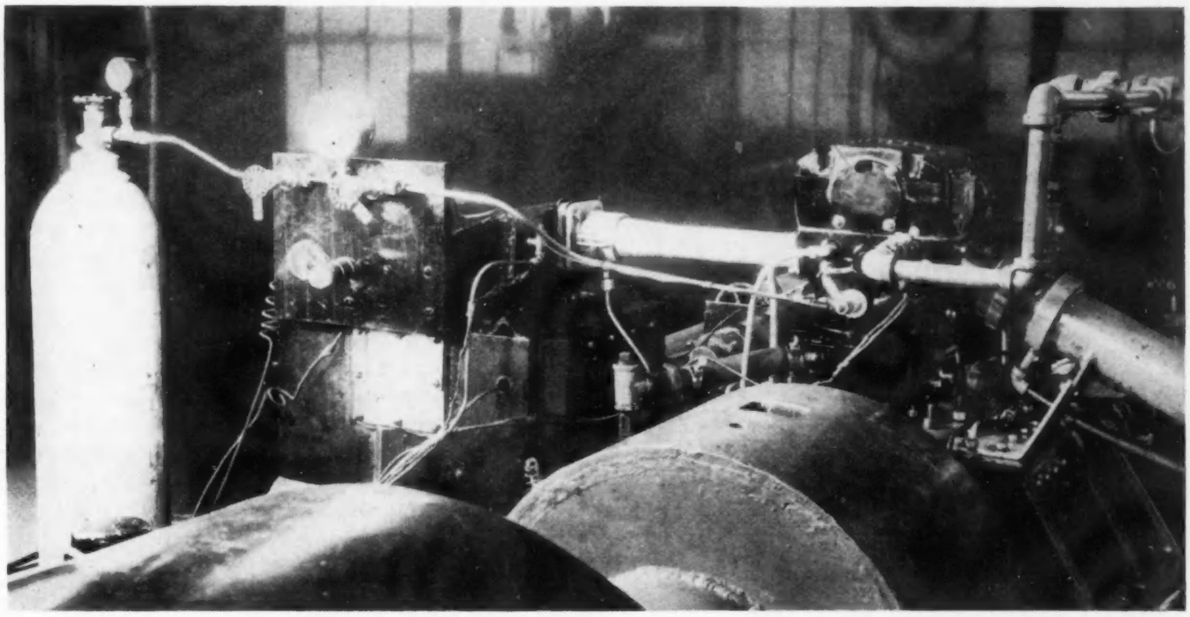


Fig. 12 – N. A. C. A. Maximum-Pressure Indicator on Engine

our supercharging apparatus at that time. This is an interesting demonstration of the lead susceptibility of iso-octane.

The effect of design differences on limiting output is shown in Fig. 15. Here a ribbed piston is compared with a piston having no cooling ribs on the underside of the head, using iso-octane as fuel. It is seen that this change reduces the allowable explosion pressure from 1530 to 1300 lb. per sq. in. The rate of increase of power with increase in manifold pressure is also slightly less. Cooling of spark-plugs, valves, and cylinder-heads is also critical factor in obtaining high allowable explosion pressures.

It is evident from an examination of Fig. 10 that, for any given maximum allowable explosion pressure, the higher outputs will be obtained at lower compression ratios. In Fig. 16, a comparison is made of limiting output on iso-octane plus 4 cc. of tetraethyl lead at 6:1 and 8:1 compression ratios. From the calculated pressures, it appears that the maximum allowable explosion pressure with this fuel and cylinder is between 1650 and 1700 lb. per sq. in. The limiting output at 6:1 is 33 per cent greater than at 8:1.

As shown in Fig. 5, the use of high manifold pressures

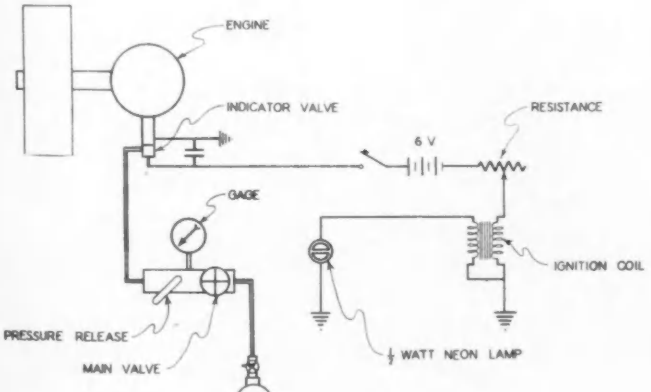


Fig. 11 – Schematic Diagram of N.A.C.A. Maximum-Pressure Indicator

results in very high manifold temperatures. The incorporation of an efficient after-cooler is desirable, both for its effect on power at any given supercharge pressure and for its effect on limiting output. We have used a method for determining the maximum desirable manifold temperature for a given cylinder design and fuel, which was suggested by the Materiel Division. The cylinder is operated at the desired indicated output, specific fuel consumption, and speed, and the manifold temperature is increased gradually. At some temperature the line of required manifold pressure vs. temperature will break upward fairly sharply as shown in Fig. 17. Here is shown a calibration of this type using 87-octane (Motor Method) aviation fuel, a compression ratio of 6:1, and indicated specific fuel consumption of 0.45 lb. per hp-hr. The break in the curve at 203 deg. fahr. indicates that it is not economical to operate under these conditions at higher manifold temperatures due to the greater pressure required from the supercharger.

A study of the foregoing section on thermal considerations of high power output immediately indicates that the mechanical design of an engine must receive appreciable attention before such cylinder pressures as quoted can be utilized fully to take advantage of the resultant powers and fuel economies.

Mechanical Considerations

It may have been noticed that, in the preceding discussion, no mention has been made of crankshaft speed. We must, therefore, now give attention to high specific output in terms of power per unit displacement rather than mean effective pressure alone. Thus the speed factor is introduced.

Analysis of available design data indicates that, with but few exceptions, an engine design based to handle high mean effective pressures will not be radically more difficult of achievement if we simultaneously increase its rotative speed above that now customary. It becomes necessary, however, to choose the speed so that the design loads will be balanced evenly. It can be shown that cylinder mean effective pressures and maximum pressures determine the size and structural shape of most component engine parts, and that a simultaneous increase in speed and mean effective pressure will result in still higher specific power outputs without serious additional weight or complications.

A most fortunate coincidence comes to the assistance of the designer. In order to protect the powerplant more fully in diving maneuvers, an over-speed factor of safety was intro-

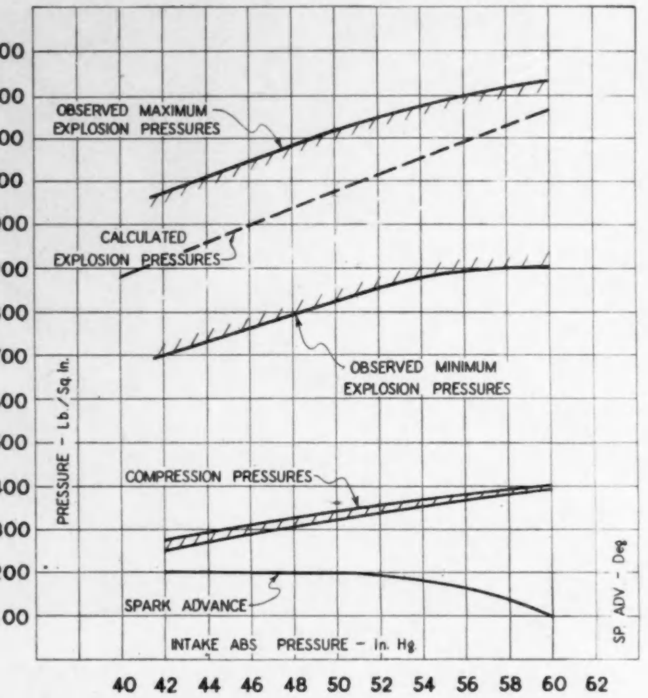


Fig. 13 – Cylinder Pressures Vs. Boost Pressures – Compression Ratio, 6:1; Fuel, Technical Iso-Octane; Air Temperature, 203 Deg. Fahr.

duced and made part of the current acceptance specifications. It had then become necessary to design an engine so that a rotative speed 20 to 30 per cent in excess of the rated speed could be maintained for short periods without disabling damage. The introduction of certain governing devices, such as the constant-speed propeller, has now so modified this picture that it is likely that future engines will be designed to permit the rated speed to approach the maximum more closely. It is quite probable, therefore, that the speed allowable for take-off or emergency maneuvers will be the highest the engine ever will be called upon to withstand.

It is not within the scope of this paper to present engine-design data but, for the purpose of carrying its presentation to a logical conclusion, a few of the major engine units and their particular design relationship to speed and pressure will be discussed.

Cylinder

The cylinder-head with its combustion-chamber is, of course, "designed" solely by maximum temperatures and pressures, and no dynamic considerations enter into its evolution. The cylinder barrel, although intended primarily as a tension member, possibly may require some checking due to the increased side thrust caused by increasing the speed. However, if designed to take care of high mean cylinder pressures, it is likely that an increase in speed will not be the determining factor.

Some question arises as to piston design. It is believed that, when a proper type has been developed for high gas pressures and severe thermal conditions with the necessary rigidity at high temperatures, it also will be suited excellently for high dynamic stresses.

Valve Gear

The valve gear is one of the few details of an engine where high operating speeds add to the stresses resulting from high gas pressures. It is true, therefore, that the valve-operating mechanism offers an obstacle to extremely high-speed engine operation.

The gear arrangement required to drive the cam-operating mechanism should not be materially different in the high-speed engine having the overhead-camshaft type of construction. The pushrod-operated valve gear may demand certain limitations at the higher speeds.

There is not the least question, however, but that the valves themselves, if adapted to high temperatures, are likewise exceedingly well adapted to extremely high-speed operation. The hollow-head valve dictated by temperature is rigid and withstands the more violent decelerations and the additional valve-head stresses caused by deflections in the valve gear at high speeds. The valve-seat and tip materials that have been evolved for operation at high temperature function admirably at high speed. The large-diameter stem necessary for adequate heat dissipation provides an excellent "cross-head" for guiding the valve with high linear velocities. Rocker side-thrust, increased

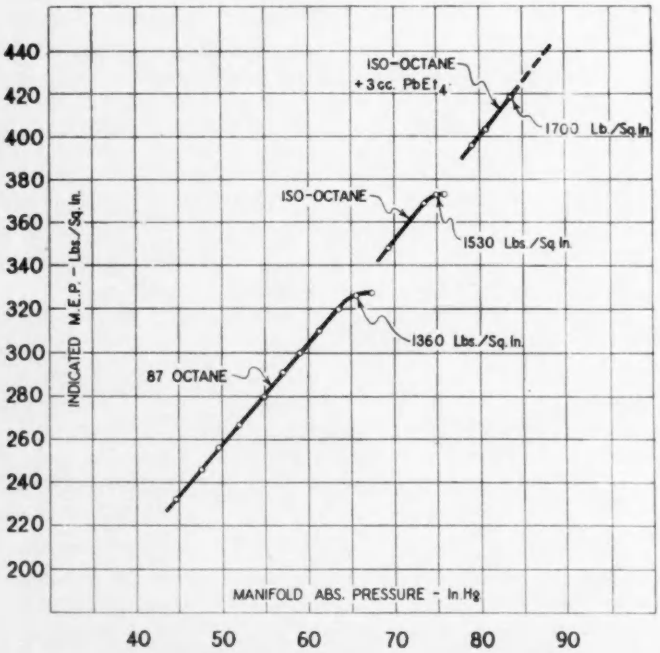


Fig. 14 – Maximum Output Calibration – Best Power Mixture; Compression Ratio, 6:1; Manifold Temperature, 200 to 210 Deg. Fahr.

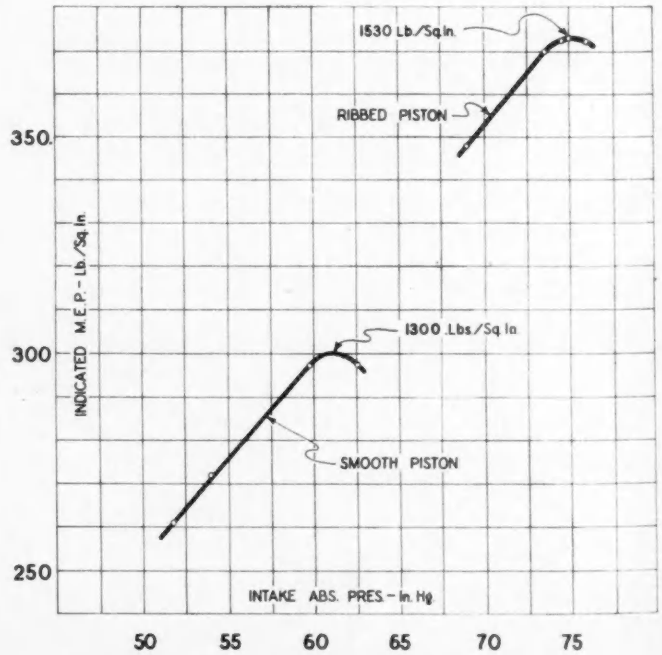


Fig. 15 – Limiting Output Vs. Design Factors – Compression Ratio, 6:1; Fuel, Iso-Octane

by the heavy springs required for high speed, also is handled better by the large stem. The designer may reduce his chances of failure by using fully lubricated valve gear. He may further take advantage of unsymmetrical cam profiles, a device which has not seen wide use so far.

Connecting-Rods

The size of the connecting-rod shank is determined solely by gas pressures in column loading. Therefore, an increase in maximum or mean pressures automatically allows the use of high speeds, as the tension loading of the rod is not likely to be the limiting design factor. The design of the crankpin end of the rod requires modification with increase in speed, mainly due to deflection of the cap and bolt.

Bearings

In conventional engine designs both connecting-rods and main bearings are designed purely by gas pressure. Thus, if

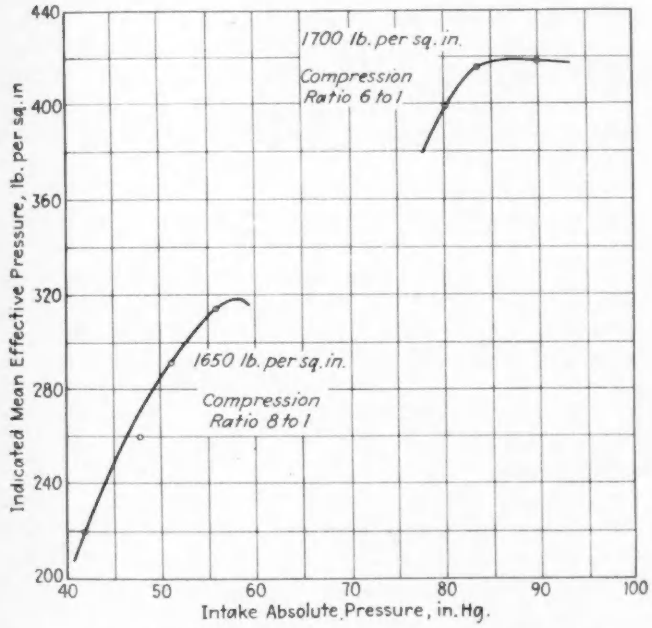


Fig. 16 – Limiting Output Vs. Compression Ratio – Fuel, Iso-Octane Plus 4 Cc. Tetraethyl Lead

the bearings will withstand high gas loadings satisfactorily, they automatically and without further difficulty will provide an engine with a set of bearings capable of further increasing the specific output by higher rotative speeds.

In order to illustrate this point, we may analyze the bearing loads of two hypothetical engines. In Fig. 18 (*A*) the main bearing loads of an engine operating at 150 lb. per sq. in. i.m.e.p. and 2000 r.p.m. have been plotted. The next diagram, Fig. 18 (*B*), shows the result of increasing the indicated mean effective pressure to 250 lb. per sq. in. The design presumably must be revamped to withstand these greatly increased loads. This second engine, however, may be speeded up to 3000 r.p.m. at the same i.m.e.p. with an actual reduction in maximum bearing loads, as shown in Fig. 18 (*C*). The mean loading, of course, is higher, but it is believed that the performance of the bearings will be better than that obtained under the loading shown in Fig. 18 (*B*), due to elimination of the "pounding effect."

A similar relation to that shown for the main bearings also holds true for the connecting-rod bearings.

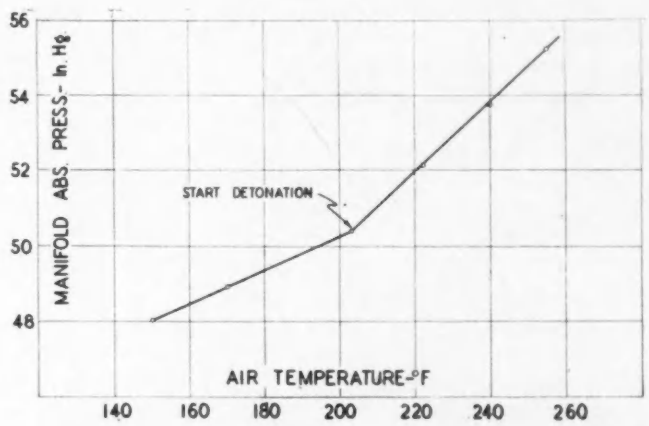


Fig. 17 – Boost Vs. Air Temperature for Constant Specific Output – Fuel, 87-Octane; Fuel Consumption, 0.45 Lb. per Hp-Hr.

It is likely that the other bearings throughout the engine never impose such design limitations that the sizes become critical, and that the majority of the engine bearings, such as those in pumps, accessories, and superchargers, have to run at specified speeds which are not involved in the rated speed of the crankshaft.

Crankshaft and Crankcase

Apart from the considerations of torsional and bending vibration resonance in the crankshaft-crankcase combination, this group is designed exclusively by gas pressures with the exception of cylinder attachments which, of course, are definitely a function of maximum cylinder pressure.

In a consideration of the vibration problem, it appears that each particular design has to be treated individually. The higher speed engine is apt to have crankcase periods set up by lower orders of excitation which would, of course, have greater amplitudes. The higher speed crankshaft will operate in the vicinity of the higher modes of vibration, but the damping will be correspondingly higher. These modes will be excited by the higher orders of excitation only and, as these higher orders are of small magnitude, the problem becomes correspondingly less difficult.[6]

Supercharger Drive

It is significant that the high impeller tip velocity required to produce the manifold pressures necessary to high mean

(*Continued on page 262*)

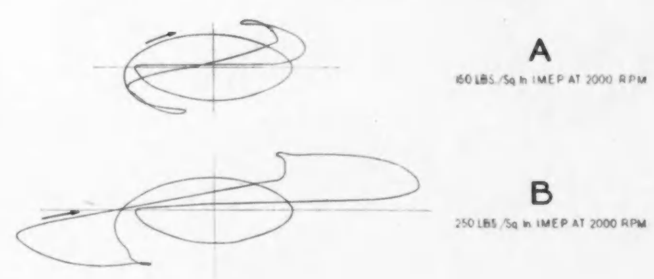


Fig. 18 – Bearing Loads Vs. Indicated Mean Effective Pressure and Revolutions per Minute

[6] See S.A.E. TRANSACTIONS, this issue, pp. 252-262; "Aircraft-Engine Installation Vibration Problems," by J. M. Tyler.

Cetane Numbers – Life Size

By Lieut.-Com. R. F. Good
U. S. Naval Engineering Experiment Station

IN January, 1934, the Bureau of Engineering of the United States Navy Department instructed the Naval Engineering Experiment Station at Annapolis, Md., to undertake a study of fuels for Diesel engines. During the preceding 25-year period in which Diesel engines in the Naval service had been of the comparatively slow-speed, air-injection type, the fuel problem had never been a serious one. With the advent of high speed and hydraulic injection in engines of much smaller cylinder dimensions, it became apparent that fuel specifications must be revised drastically if engine performance was to be maintained at the high standard required for Naval effectiveness.

The preparation of satisfactory specifications has required consideration of practically every defined property of petroleum derivatives, the selection of those properties having major significance with respect to the fundamental requirements for a good fuel and, in some cases, the development of test procedures for evaluation of the selected properties.

The fundamental requirements of a fuel acceptable for Naval use were described broadly as:

(1) A single grade of fuel suitable for all purposes (except aviation) must be specified. This requirement is dictated by limited tankage accommodations on board ship and by simplification of supply and distribution.

(2) The selected fuel must be safe and stable in storage, and non-corrosive to materials used for tanks and piping. It must be capable of being handled at winter temperatures without preheating.

(3) It must insure a high order of engine performance with respect to starting, specific fuel consumption, smokeless combustion, and smooth running under both summer and winter weather conditions.

(4) It must help to minimize outage for cleaning and repair; that is, it must not cause cylinder or nozzle deposits or sludging of the lubricating oil, it must not contain corrosive or abrasive impurities, and must not contribute to mechanical failure by repeated severe combustion shock.

(5) The selected fuel must be capable of production by ordinary refinery processes from a wide variety of crudes in order that availability of supply, particularly in times of National emergency, may be insured.

(6) It should be capable of production at non-premium prices, unless the military advantages afforded are so great as to outweigh all cost considerations.

This paper presents the results of a study of Diesel-fuel ignition quality with respect to engine performance and maintenance, items (3) and (4) preceding. The introductory paragraphs are intended only to show the relationship which this one phase of the subject bears to the complete investigation.

Materials Under Test

More than 100 fuels have been tested in from 2 to 4 different types of engines for periods ranging from 8 to over 100 hr. each. For ready identification, the fuels were assigned to alphabetical groups.

The test fuels in the *A* and *B* groups were obtained in small quantities by random purchase. Fuel groups *C* to *G* inclusive were purchased in quantity to special order. Each group contained five samples: Fuel 1, a straight-run distillate in the gas-oil fraction; Fuel 2, a cracked product from the same refinery run; and Fuels 3 to 5, inclusive, 75/25, 50/50, and 25/75 per cent blends, respectively, of Fuels 1 and 2. The fuels in Group *C* were derived from a mixed Mid-Continent crude, Groups *D* and *G* from two West-Coast fields, Group *E* from a mixed Mid-Continent/West-Texas blend, and Group *F* from a Pennsylvania crude. Fuel *H*-1 is a single sample prepared by the hydrogenation process from a Louisiana source. Station Fuel is representative of contract deliveries, stocked in quantity at the Experiment Station for powerplant and motorboat use. Fuels 4 and 5 are from earlier contract deliveries, aged three years before testing, and Fuel *M*-306 is a straight-run Mid-Continent refined to aviation specifications.

In addition to testing the fuels as received, two series of cross blends, one composed of Fuels *B*-11 and *B*-12, the second of Fuels *F*-1 and *G*-2, were prepared. These series were chosen because, although each gave a wide range of ignition quality, they represented, respectively, the upper and lower limits of viscosity.

The cross-blended series and Fuel *E*-2 also were doped with varying percentages of ethyl nitrate to determine the effect of this compound in reducing ignition delay. Finally, the *E*-2 sample containing 2 per cent of ethyl nitrate was given a secondary infusion of commercial ethyl fluid in varying amounts in an effort to ascertain whether or not the burning rate could be suppressed concurrently with reduced ignition delay.

Data are tabulated in Figs. 1 and 2. For primary and secondary reference fuels, the recommendations of the Volunteer Group for Compression-Ignition Research[1] were followed. Pure cetane was used as the high-ignition-quality primary and alpha-methylnaphthalene as the low-ignition primary. Shell high-cetane reference fuel and the commercial grade of mixed alpha and beta-methylnaphthalenes were used as the secondary reference fuels.

Apparatus, Equipment, Test Methods

The usual laboratory equipment was available for the determination of the physical and chemical properties of test and reference fuels.

From the evaluated physical and chemical properties, three ignition-quality indexes were computed: Diesel index number[2],

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 13, 1937.]

[1] See S.A.E. TRANSACTIONS, June, 1935, pp. 206-209; "Progress Report of Volunteer Group for Compression-Ignition Fuel Research," by T. B. Rendel.

[2] See S.A.E. TRANSACTIONS, October, 1934, pp. 376-384; "A Suggested Index of Diesel Fuel Performance," by A. E. Becker and H. G. M. Fischer.

viscosity-gravity constant[3], and boiling point-gravity constant[4], in order that the value of these indexes relative to those obtained from engine tests might be shown.

Fig. 1 – Physical and Chemical Properties of Test Fuels

Test	Method: F.S.B. No.	Method: A.S.T.M. No.	F-1	F-1 + 1% EN	F-1 + 3% EN	F-1 + 5% EN	80%F-1 20%G-2 +1% EN	60%F-1 40%G-2 +1% EN	40%F-1 60%G-2 +1% EN	40%F-1 60%G-2 +3% EN	40%F-1 60%G-2 +5% EN	20%F-1 80%G-2 +1% EN	G-2	G-2 + 1% EN	G-2 + 3% EN	G-2 + 5% EN	E-2	E-2 + 1% EN	E-2 + 3% EN	E-2 + 5% EN	E-2 + 2% EN + 1cc EF	E-2 + 2% EN + 3 cc EF	E-2 + 2% EN + 6 cc EF	Station Fuel
Specific gravity at 60/60 deg.F.	Pycnometer		0.825	0.829	0.836	0.843	0.846	0.857	0.880	0.887	0.893	0.897	0.911	0.914	0.920	0.925	0.880	0.883	0.889	0.898	0.886*	0.886*	0.886*	0.868
A.P.I. gravity at 60/60 deg.F.	a		40.02	39.18	37.76	36.36	35.76	33.62	29.00	28.02	26.95	26.25	23.82	23.32	22.30	21.47	29.30	28.75	27.67	26.42	28.20*	28.20*	28.20*	32.84
Flash point, closed cup, deg.F.	110.22	D93-22	206	149	133	124	149	149	149	133	133	149	176	128	114	106	194	141	125	116	131	131	131	200
Pour point, deg.F.	20.14	D97-33	+20	+20	+20	+20	+20	+20	+15[c]	+15[c]	+10[c]	0[c]	Below -40	Below -40	Below -40	Below -40	+5	+5	+5	+5	+5*	+5*	+5*	+5
Viscosity, SSU, at 100 deg.F.	30.43	D88-33	41.0	40.7	40.0	39.4	40.2	39.5	39.0	38.5	38.0	38.3	38.0	37.8	37.4	37.0	42.0	41.7	41.0	40.3	41.4*	41.4*	41.4*	39
Total sulphur, % by weight	[b]520.22	[b]D129-33	0.11	0.11	0.11	0.10	0.18	0.25	0.31	0.30	0.29	0.38	0.45	0.45	0.44	0.43	0.89	0.88	0.86	0.85	0.87	0.87	0.87	0.40
Water & Sediment, % by volume	300.32	D96-30	0	0	0	0	0.22	0.44	0.66	0.65	0.64	0.88	1.10	1.09	1.07	1.05	0.01	0.01	0.01	0.01	0.01	0.01	0.01	Trace
Carbon residue (Conradson), % by wt	500.13	D189-30	0.003	0.003	0.003	0.003	0.041	0.079	0.114	0.111	0.108	0.153	0.192	0.190	0.186	0.182	0.432	0.428	0.419	0.410	0.423	0.423	0.423	0.033
Ash, % by weight	542.1	-	0	0	0	0	0.007	0.013	0.020	0.020	0.020	0.026	0.033	0.033	0.032	0.031	Trace	Trace	Trace	Trace	Trace	Trace	Trace	0
Distillation Range, deg.F.	100.14	D86-30																						
Initial Boiling Point			442	190	190	190	190	190	190	190	190	190	385	190	190	190	386	190	190	190	190	190	190	428
10%			503	500	495	490	479	462	450	443	435	442	440	437	430	422	447	442	434	427	438	438	438	475
50%			548	547	545	543	541	534	524	521	518	510	498	497	494	491	535	533	529	525	531	531	531	552
90%			637	632	621	613	628	617	611	605	598	604	604	598	587	578	685	674	657	643	664	664	664	657
End point			705	705	705	705	700+	700+	700+	700+	700+	700+	692	692	692	692	700+	700+	700+	700+	700+	700+	700+	719
Aniline point, deg.F.	310.11	-	182.8	180.7	177.3	175.0	164.8	147.0	129.0[d]	126.6[d]	124.3[d]	109.0[d]	83.8[d]	83.6[d]	81.6[d]	79.8[d]	137.5[d]	134.8[d]	132.8[d]	130.4[d]	131.3[d]	133.5[d]	134.6[d]	144.1

a. From specific gravity and National Standard Petroleum Oil Tables, Bureau Standards No.154.
b. Using bomb washings from heat value determination.
c. Estimated.
d. Approximate, due to dark color of oil.

EN represents ethyl nitrate, $C_2H_5ONO_2$, blended % by volume.

EF represents ethyl fluid, 68% $Pb(C_2H_5)_4$, blended cc per U.S. gallon.

* Neglecting effect of ethyl fluid.

The engines used for the tests reported in this paper were:

(1) The standard C.F.R.-A.S.T.M. Diesel knock-testing engine, equipped for ignition-quality ratings by either the Critical-Compression-Ratio[5] or Knockmeter-Delay[6] methods. The engine was later fitted for determinations by the Fixed-Ignition-Delay[7] method. Blending-Octane-Number[8] ratings were made in a second C.F.R. engine. Since, however, the Volunteer Group for Compression-Ignition Research has implied tentative acceptance of the first two methods, only those results are reported in Fig. 2.

(2) A special single-cylinder test unit, representative of, and using the standard operating parts of, a submarine main propelling engine. This engine is rated 75 b.hp. at 750 r.p.m., 8-in. bore by 10-in. stroke, two-cycle, solid-injection, open-type combustion-chamber. Fig. 3 shows a three-quarter view of this engine as arranged to drive a direct-current, cradle-frame dynamometer. All full-scale testing reported in this paper was done with this engine.

Ignition-delay measurements were made from simultaneous recordings of crankshaft angular velocity, start of injection, and start of cylinder-pressure rise following ignition, all obtained with a magnetic-type, two-element oscillograph. Cyclic events in the test engine may be viewed continuously with this instrument. For permanence and to permit accurate mensuration, photographic reproductions were made at predetermined intervals.

The time axis of the oscillograph is produced by an oscillating mirror operated in synchronism with the test unit. Phase location of cyclic events with respect to crankshaft angular position was accomplished by mounting a permanent magnet surrounded by an induction coil adjacent to the path of the flywheel bolt heads. As each bolt head passes the magnet a small instantaneous electromotive force is induced in the coil. The two bolt heads nearest firing center were turned down to sharpen the timing impulses induced by them and so distinguish them from the other six. The coil was mounted so that these impulses occurred 32 deg. before and 13 deg. after firing center, in order that the injector-valve motion, recorded on the same galvanometer, would not be partly obscured.

The exact start of injection could not be recorded directly as it was not practicable to insulate the injector valve from its seat. The arrangement adopted was such that the valve closed an electrical contact after it had moved 0.013 in. Energy for this circuit was obtained from a 3-volt dry battery. Details of the various circuits are given in Fig. 4.

The point-of-ignition pressure-rise indicator developed for this test also is illustrated in Fig. 4. The motion of a diaphragm in communication with the engine combustion-chamber varies the air gap of a permanent magnet surrounded by an induction coil. The diaphragm used consists of two hardened alloy-steel laminations, each 2 in. in diameter by 0.050 in. thick. The magnet pole piece and diaphragm cage are of hardened tool steel. The cage is copper-jacketed for water-cooling. The coil consists of 3200 turns of No. 31 magnet wire and has a resistance of 120 ohms. The unit is magnetized initially by momen-

[3] See *Industrial and Engineering Chemistry*, June, 1928, pp. 641-644; "The Viscosity-Gravity Constant of Petroleum Lubricating Oils," by J. B. Hill and H. B. Coats, presented at the 74th Meeting of the American Chemical Society, Detroit, Mich., September, 1927, under the title: "Relation between Viscosity and Gravity of Petroleum Products."

[4] See *The Oil and Gas Journal*, March 21, 1935, pp. 16 and 20; "Boiling Point-Gravity Constant Is Index of Lubricating-Oil Characteristics," by E. A. Jackson.

[5] See S.A.E. Transactions, March, 1932, pp. 136-142; "Compression-Ignition Characteristics of Injection-Engine Fuels," by A. W. Pope, Jr., and J. A. Murdock.

[6] See "Tentative Standard Operating Conditions and Procedure for Delay Method of Rating Diesel Fuels on Waukesha High-Turbulence Diesel," Waukesha Motor Co., July, 1935.

[7] See *Automotive Industries*, Aug. 17, 1935, pp. 202-205; "A New Technique and Apparatus for Testing Diesel Fuels," by P. H. Schweitzer and T. B. Hetzel.

[8] See *National Petroleum News*, Jan. 22, 1936, pp. 22-28, and Jan. 29, 1936, pp. 25-28; "The Effect of Crude Source on Diesel Fuel Quality," by W. H. Hubner and G. B. Murphy.

Submarine Type Engine, CFR Engine and Physical-Chemical Fuel Performance and Ignition Quality Data for Undoped, Doped, Reference and Cetane Fuels

	Submarine Type Engine Ignition & Combustion Performance										CFR Engine Indices							Physical-Chemical Indices		
Fuel Designation	Date of Tests	Ignition Delay	Maximum Cylinder Pressure	Explosion Pressure Rise	Burning Rate	Computed Combustion Knock	Ignition Delay Ref. Fuel No.	Ignition Delay Cetane No.	Combustion Knock Ref. Fuel No.	Combustion Knock Cetane No.	Date of C.C.R. Tests	Critical Compression Ratio	C.C.R. Method Ref. Fuel No.	C.C.R. Method Cetane No.	Date of Knockmeter Delay Tests	Knockmeter Delay Ref. Fuel No.	Knockmeter Delay Cetane No.	Diesel Index	Boiling Point Gravity Constant	Viscosity Gravity Constant
Station Fuel	Sept.1	9.38	1002	175	42	.73	73.8	55.9	64.6	50.6	(3)	(3)		(3)	(3)		(3)	47.32	186.5	0.850
C.1	"	9.30	1010	169	37	.62	75.0	56.9	73.7	58.3	(8)	8.36		42.6	(6)		47.6	61.09	180.8	0.819
C.3	"	9.50	1019	180	41	.74	71.7	54.3	64.2	50.1	"	8.28		44.8	"		45.8	57.00	182.1	0.826
C.4	"	9.51	1022	185	43	.79	71.6	54.2	61.2	47.6	"	8.51		41.6	"		44.2	53.42	183.4	0.828
C.5	"	9.67	1039	187	44	.82	69.0	52.2	59.8	46.3	"	8.51		40.6	"		41.3	49.84	184.9	0.835
C.2	"	10.00	1028	195	44	.86	63.8	48.0	58.0	44.7	"	9.01		35.7	"		39.3	45.61	186.8	0.840
Station Fuel	"	9.51	1010	175	40	.70	71.6	54.2	66.7	52.3	(3)	(3)		(3)	(3)		(3)	47.32	186.5	0.850
100% High Reference Fuel	"	8.00	1004	155	30	.47		76.9		79.8	(7)	6.91		72.3	(6)		72.0	73.23	173.2	0.806
90% " " "	"	8.68	1012	164	32	.53		68.9		71.8	"	7.02		65.6	"		64.8	61.27	179.7	0.828
75% " " "	"	9.40	989	169	37	.62		56.9		59.3	"	7.50		54.7	"		54.5	43.27	190.2	0.862
60% " " "	"	10.13	978	182	41	.75		45.0		46.6	"	8.33		42.6	"		44.2	29.51	201.3	0.894
45% " " "	"	11.49	1015	239	57	1.36		33.2		33.3	"	9.90		30.8	"		34.0	18.05	209.6	0.927
25% " " "	"	12.81	1007	287	66	1.88		25.2		25.2	"	11.50		24.3	"		27.0	11.32	214.9	0.950
Station Fuel	"	9.78	978	179	42	.75	67.3	50.8	63.3	49.4	(3)	(3)		(3)	(3)		(3)	47.32	186.5	0.850
D.1	"	10.15	997	183	44	.81	61.6	46.3	60.1	46.6	(8)	8.58		40.2	(6)		41.4	44.53	187.8	0.852
D.3	"	10.21	1005	195	44	.87	60.8	45.7	57.3	44.2	"	8.50		41.6	"		41.1	43.98	188.2	0.852
D.4	Oct.12	10.42	996	190	45	.86	61.9	46.5	51.6	39.2	"	8.51		41.0	"		41.5	43.18	188.7	0.854
D.5	Sept.1	10.33	1001	195	45	.88	59.2	44.4	56.8	43.7	"	8.55		40.9	"		42.3	42.11	189.6	0.856
D.2	"	10.33	1010	198	45	.90	59.2	44.4	55.8	42.8	"	8.53		40.4	"		41.7	41.48	189.7	0.856
Station Fuel	"	9.74	998	179	39	.70	67.7	51.2	66.7	52.2	(3)	(3)		(3)	(3)		(3)	47.32	186.5	0.850
Station Fuel	Oct.12	10.07	1005	167	40	.66	67.2	50.7	61.0	47.3	(3)	(3)		(3)	(3)		(3)	47.32	186.5	0.850
E.1	Aug.26	9.43	1010	175	40	.70	(1)		(1)		(8)	7.93		47.5	(6)		47.1	50.60	184.3	0.842
E.3	Oct.12	9.90	984	154	37	.58	69.8	52.7	67.0	52.4	"	7.97		46.6	"		46.7	47.28	186.2	0.846
E.4	"	10.13	985	166	42	.69	66.3	50.0	59.3	45.9	"	8.00		45.6	"		44.3	44.82	187.9	0.854
E.5	"	10.23	983	182	41	.74	64.8	48.8	57.1	44.0	"	8.15		43.9	"		42.3	42.50	189.1	0.855
E.2	"	10.47	988	180	43	.77	61.4	46.1	55.5	42.5	"	8.28		42.8	"		42.1	40.29	190.7	0.865
Station Fuel	"	9.90	1010	172	40	.70	69.9	52.7	58.8	45.3	(3)	(3)		(3)	(3)		(3)	47.32	186.5	0.850
100% High Reference Fuel	"	8.50	969	138	30	.42		77.0		79.8	(7)	6.91		72.3	(6)		72.0	73.23	173.2	0.806
90% " " "	"	8.89	984	136	31	.43		69.0		72.0	"	7.02		65.6	"		64.8	61.27	179.7	0.828
75% " " "	"	9.62	985	154	34	.53		57.0		59.4	"	7.50		54.7	"		54.5	43.27	190.2	0.862
60% " " "	"	10.44	972	169	41	.69		45.0		46.2	"	8.33		42.6	"		44.2	29.51	201.3	0.894
45% " " "	"	11.99	982	192	52	.99		33.2		33.6	"	9.90		30.8	"		34.0	18.05	209.6	0.927
35% " " "	"	13.74	1024	274	70	1.92		25.3		25.7	"	11.50		24.3	"		27.0	11.32	214.9	0.950
Station Fuel	Sept.2	9.59	977	169	39	.66	65.7	49.6	66.4	52.0	(3)	(3)		(3)	(3)		(3)	47.32	186.5	0.850
G.1	"	10.00	995	195	46	.90	59.6	44.7	52.7	40.1	(8)	8.78		38.1	(6)		40.0	44.15	189.3	0.854
G.3	"	10.63	978	197	49	.96	51.5	38.3	50.3	38.2	"	8.37		37.3	"		36.1	37.21	192.7	0.869
G.4	Oct.12	11.02	1028	179	53	.95	45.8	33.8	49.2	37.2	"	9.43		33.5	"		32.2	30.77	196.7	0.881
G.5	Sept.2	11.72	991	251	60	1.51	40.4	29.5	37.2	27.2	"	10.14		29.9	"		31.2	25.25	199.1	0.891
G.2	"	12.27	999	272	67	1.82	36.2	26.2	33.7	24.3	"	10.30		27.6	"		28.6	20.06	201.8	0.904
Station Fuel	"	9.64	975	170	42	.71	65.0	49.0	62.8	49.0	(3)	(3)		(3)	(3)		(3)	47.32	186.5	0.850
60% High Reference Fuel	"	9.77	984	181	40	.72		45.0		46.6	(7)	8.33		42.6	(6)		44.2	29.51	201.3	0.894
45% " " "	"	11.31	956	212	51	1.09		33.2		33.3	"	9.90		30.8	"		34.0	18.05	209.6	0.927
35% " " "		12.30	993	268	62	1.66		25.2		25.2	"	11.50		24.3	"		27.0	11.32	214.9	0.950
Station Fuel	Sept.3	9.69	984	172	38	.65	73.3	55.6	69.8	54.8	Oct.20	7.93	72.0	52.6	Oct.27	61.4	44.4	47.32	186.5	0.850
100% F.1	"	8.97	1001	164	33	.54	86.2	65.9	79.9	63.3	"	6.96	100.0	74.7	Oct.29	88.3	67.5	73.36	174.9	0.807
80% F.1 + 20% G.2	"	9.23	994	177	38	.66	81.2	61.8	68.8	54.0	"	7.30	84.2	62.0	Nov.2	76.4	57.3	61.04	180.1	0.827
60% F.1 + 40% G.2	"	9.85	983	182	38	.69	70.4	53.3	66.7	52.3	"	7.93	72.0	52.6	"	59.7	42.8	49.08	185.0	0.846
40% F.1 + 60% G.2	"	10.23	1005	191	43	.81	64.3	48.7	59.7	46.3	"	8.86	59.9	43.3	"	53.5	37.7	38.32	190.7	0.867
20% F.1 + 80% G.2	"	11.17	1004	223	52	1.15	51.5	38.3	47.5	35.6	"	9.34	50.2	35.9	"	43.2	28.9	28.90	196.1	0.886
100% G.2	"	12.44	1023	277	64	1.77	38.3	27.9	36.3	26.4	"	11.23	40.8	28.7	Oct.30	36.6	23.3	19.39	201.8	0.904
Station Fuel	"	9.85	1007	169	40	.67	70.4	53.3	68.0	53.2	" Oct.19	7.98 8.04	69.1	50.4	Oct.27	61.4	44.4	47.32	186.5	0.850
100% High Reference Fuel	"	8.56	940	150	31	.46		76.9		79.8	" "	6.99 6.90		74.7	(9)		77.0	73.23	173.2	0.806
90% " " "	"	8.76	980	150	33	.50		68.9		71.8	" "	7.17 7.14		66.6	"		69.2	61.27	179.7	0.828
75% " " "	"	9.59	976	163	36	.58		56.9		59.3	" "	7.42 7.43		54.9	"		56.1	43.27	190.2	0.862
60% " " "	"	9.99	1008	188	43	.80		45.0		46.6	" "	9.04 8.65		43.4	"		43.2	29.51	201.3	0.894
45% " " "	"	11.71	964	203	48	.98		33.2		33.3	" "	10.49 10.49		31.8	"		30.3	18.05	209.6	0.927
35% " " "	"	12.82	1017	291	69	2.00		25.2		25.2	" "	12.59 12.36		24.1	"		22.1	11.32	214.9	0.950
Station Fuel	"	9.81	995	169	41	.70	71.3	53.9	66.0	51.7	" "	7.93 7.98	70.0	51.0	Oct.27	61.4	44.4	47.32	186.5	0.850
100% B.11	"	8.61	983	157	36	.56	93.4	71.6	77.7	61.6	"	7.42	78.0	57.1	"	81.0	61.2	57.18	180.9	0.834
75% B.11 + 25% B.12	"	8.98	1004	163	37	.60	86.0	65.8	73.2	58.0	"	7.77	71.2	52.1	"	73.2	54.5	52.23	182.9	0.844
50% B.11 + 50% B.12	"	9.13	996	171	36	.62	83.0	63.3	71.9	56.7	"	8.13	65.8	47.8	Oct.27 Nov.16	67.6	49.6	47.16	184.9	0.854
25% B.11 + 75% B.12	"	9.63	1001	179	41	.73	73.8	56.0	64.3	50.3	"	8.47	61.9	44.9	Oct.27	63.1	45.9	42.39	187.3	0.861
100% B.12	"	10.08	1023	196	45	.88	67.0	50.5	56.7	43.7	"	8.76	58.9	42.4	Oct.29	59.7	42.9	37.73	190.9	0.871
Station Fuel	"	9.81	936	165	39	.65	71.2	53.9	69.3	54.8	"	8.04	67.1	48.9	Oct.27	61.4	44.4	47.32	186.5	0.850
Station Fuel	Sept.4	9.79	938	170	38	.65	64.6	48.7	71.8	56.7	(3)	(3)		(3)	(3)		(3)	47.32	186.5	0.850
H.1	"	9.23	950	158	34	.54	73.3	55.5	83.0	66.1	(8)	7.86		49.2	(6)		47.6	62.04	180.1	0.829
M.306	"	9.29	954	170	35	.60	72.4	54.9	75.7	59.9	Sept.11	9.00		36.2	Sept.8 Oct. 8	65.6	48.0	61.74	180.5	0.834
Fuel Change-Over Calibration	"																			
Station Fuel	"	(2) 10.00	974	185	41	.75	61.2	46.0	63.8	49.8	"	(11)7.71	68.9	50.1	Oct.27	61.4	44.4	47.32	186.5	0.850
100% High Reference Fuel	"	7.89	993	146	32	.46		76.9		79.8	"	7.33		74.7	(9)		77.0	73.23	173.2	0.806
90% " " "	"	8.55	994	154	34	.53		68.9		71.8	"	7.77		66.6	"		69.2	61.27	179.7	0.828
75% " " "	"	9.39	1016	169	36	.61		56.9		59.3	"	(4) 8.23		54.9	"		56.1	43.27	190.2	0.862
60% " " "	"	9.85	1013	197	40	.79		45.0		46.6	"	9.20		43.4	"		43.2	29.51	201.3	0.894
45% " " "	"	10.87	1006	221	52	1.15		33.2		33.3	"	(4)10.97		31.8	"		30.3	18.05	209.6	0.927
35% " " "	"	12.51	988	250	60	1.50		25.2		25.2	"	(4)12.77		24.1	"		22.1	11.32	214.9	0.950
Station Fuel	"	9.55	981	167	38	.64	68.2	51.5	72.6	57.3	"	(11)7.78	67.7	49.2	Oct.27	61.4	44.4	47.32	186.5	0.850
F.1	"	8.23	1006	149	34	.51	93.5	71.8	87.8	70.0	"	" 7.08	84.1	62.0	Sept11	88.3	67.5	72.92	174.9	0.807
F.3	"	8.67	973	156	35	.55	83.9	64.0	82.0	65.2	"	" 8.00	63.7	46.2	"		(10)	49.85	186.0	0.849
F.4	"	9.10	1016	188	42	.79	75.7	57.4	61.3	47.7	"	" 8.64	54.6	39.2	"	-	46.9	34.81	194.0	0.888
F.5	"	10.63	1016	223	52	1.16	52.8	39.3	45.7	34.1	"	" 10.19	41.3	29.2	"		(10)	(5)26.80	200.0	0.934
Station Fuel	"	9.60	992	179	37	.67	67.4	50.9	69.7	54.8	"	" 7.75	48.1	49.7	Oct.27	61.4	44.4	47.32	186.5	0.850

NOTES:-

(1) Submarine engine performance data from 100-hour test. Reference fuel and cetane data not available.

(2) Taken from Fuel Change-Over Calibration.

(3) Station fuel not tested as a check fuel with these test fuels.

(4) Taken from calibration curve of 100%, 90%, 80%, 60%, 40% and 30% reference fuel - CCR Tests.

(5) Extrapolated from curve for F.1, F.3, F.4 and F.5.

(6) Taken from Report Serial Number EES-6395-D and subsequent data.

Fig. 2

tarily applying 230 volts direct current to the coil terminals. In use, the coil is short-circuited across the second oscillograph galvanometer. By varying the relative inductance and capacity of this circuit, the resulting diagram may be made to represent pressure, rate-of-pressure-change, or an intermediate combination of the two. For illustrative purposes, the diagrams selected approximate the conventional pressure-time curve. Representative oscillograms are given in Fig. 5. Because of the nature of the diagram used, the oscillograms could not be calibrated for pressure measurements. They have been used solely to determine the ignition delay in crank-angle degrees. Dependence for pressure measurements was placed in a standard mechanical indicator of the cantilever-beam-spring type. Representative indicator cards also are given in Fig. 5. Enlarged indicator cards and oscillograms, with the methods of measurements indicated, are shown in Fig. 6.

As determined, the ignition-delay values are in error from two causes. The apparent delay is shorter than the true delay by the time necessary for the injector valve to rise 0.013 in. from its seat and close the injector contact maker. Because of the necessity for water-cooling the pickup unit, it could not be mounted with the diaphragm in the combustion-chamber boundary, but was placed at the end of the gas passage normally connecting to the cylinder-relief valve. Apparent delay is, there-

Submarine Type Engine, CFR Engine and Physical-Chemical Fuel Performance and Ignition Quality Data for Undoped, Doped, Reference and Cetane Fuels

Fuel Designation	Submarine Type Engine Ignition & Combustion Performance										CFR Engine Indices							Physical-Chemical Indices		
	Date of Tests	Ignition Delay	Maximum Cylinder Pressure	Explosion Pressure Rise	Burning Rate	Computed Combustion Knock	Ignition Delay		Combustion Knock		Date of C.C.R. Tests	Critical Compression Ratio	C.C.R. Method		Date of Knockmeter Delay Tests	Knockmeter Delay		Diesel Index	Boiling Point Gravity Constant	Viscosity Gravity Constant
							Ref. Fuel No.	Cetane No.	Ref. Fuel No.	Cetane No.			Ref. Fuel No.	Cetane No.		Ref. Fuel No.	Cetane No.			
Station Fuel	Oct.13	10.39	1005	154	37	0.58	67.5	51.0	66.5	51.5	Oct.19	8.10	66.3	48.2	Oct.27	61.4	44.4	47.32	186.5	.850
E.2	"	10.72	989	177	42	0.75	62.3	46.8	55.4	42.3	"	8.65	60.0	43.4	Oct.30	58.0	41.3	40.3	190.5	.864
E.2 + 1% $C_2H_5NO_3$	"	9.86	980	132	31	0.41	75.1	57.0	90.0	72.0	"	8.47	61.9	44.9	Nov.5	73.6	54.8	39.0	190.9	.868
E.2 + 3% $C_2H_5NO_3$	"	8.57	988	126	28	0.36	97.0	74.0	103.0	82.0	"	7.92	69.0	50.1	"	84.6	64.4	36.8	193.5	.877
E.2 + 5% $C_2H_5NO_3$	"	8.12	1007	128	27	0.35	106.5	82.8	106.5	84.5	"	7.68	73.0	53.2	"	93.9	72.2	34.4	195.7	.885
E.2	"	10.33	985	170	40	0.68	68.0	51.5	59.0	44.8	"	8.65	60.0	43.4	Oct.30	58.0	41.3	40.3	190.5	.864
Station Fuel	"	10.51	976	156	37	0.58	65.5	49.7	66.5	51.5	" Oct.21	8.05 7.92	67.0 68.9	48.8 50.1	Oct.27	61.4	44.4	47.32	186.5	.850
100% High Reference Fuel	"	8.37	978	124	30	0.37		77.0		79.8	" "	6.90 6.93		74.7	(9)		77.0	73.23	173.2	.806
90% " " "	"	9.52	1007	140	32	0.44		69.0		72.0	" "	7.14 7.05		66.6	"		69.2	61.27	179.7	.828
75% " " "	"	9.89	978	145	35	0.50		57.0		59.4	" "	7.43 7.36		54.9	"		56.1	43.27	190.2	.862
60% " " "	"	10.84	1003	165	40	0.66		45.0		46.2	" "	8.65 8.67		43.4	"		43.2	29.51	201.3	.894
45% " " "	"	12.06	1007	199	51	1.02		33.2		33.6	" "	10.49 10.37		31.8	"		30.3	18.05	209.6	.927
35% " " "	"	13.47	1001	268	69	1.84		25.3		25.7	" "	12.36 12.19		24.1	"		22.1	11.32	214.9	.950
Station Fuel	"	10.45	982	153	38	0.59	66.3	50.0	66.0	51.4	" "	8.10 7.84	66.3 69.9	48.2 50.8	Oct.27	61.4	44.4	47.32	186.5	.850
E.2	"	10.46	976	165	42	0.69	66.0	49.7	58.5	45.0	"	8.46	62.3	45.2	Oct.30	58.0	41.3	40.3	190.5	.864
E.2 + 2% $C_2H_5NO_3$ + 1 cc $(C_2H_5)_4Pb$..	"	9.27	983	143	31	0.44	84.7	64.7	82.5	66.0	"	8.00	67.8	49.3	Nov.6	74.0	55.1	37.0	193.8	.873
E.2 + 2% $C_2H_5NO_3$ + 3 cc $(C_2H_5)_4Pb$..	"	9.65	967	141	31	0.44	78.8	60.0	82.5	66.0	"	8.04	67.1	48.9	"	70.1	51.9	37.7	193.8	.873
E.2 + 2% $C_2H_5NO_3$ + 6 cc $(C_2H_5)_4Pb$..	"	9.82	985	152	32	0.48	75.8	57.5	77.7	61.7	"	8.23	64.9	47.1	"	59.3	42.6	38.0	193.8	.873
E.2	"	10.48	961	167	38	0.63	66.0	49.8	63.4	49.5	"	8.46	62.3	45.2	Oct.30	58.0	41.3	40.3	190.5	.864
Station Fuel	"	10.15	985	159	39	0.62	71.0	53.5	63.3	49.4	"	7.91	69.1	50.2	Oct.27	61.4	44.4	47.32	186.5	.850
Station Fuel	Oct.14	10.72	979	158	41	0.64	62.3	47.0	63.5	49.6	Oct.20	7.93	72.0	52.6	Oct.27	61.4	44.4	47.32	186.5	.850
100% F.1	"	9.84	1003	118	32	0.38	77.0	58.6	90.7	73.0	"	6.96	100.0	74.7	Oct.29	88.3	67.6	73.1	174.9	.807
80% F.1 + 20% G.2	"	10.13	963	128	33	0.42	72.0	54.6	84.8	67.7	"	7.30	84.2	62.0	Nov.2	76.4	57.3	61.04	180.1	.827
60% F.1 + 40% G.2	"	10.39	989	148	35	0.52	67.8	51.0	74.7	59.0	"	7.93	72.0	52.6	"	59.7	42.8	49.08	185.0	.846
40% F.1 + 60% G.2	"	11.44	998	163	40	0.65	52.6	39.2	63.2	49.2	"	8.86	59.9	43.3	"	53.5	37.7	38.32	190.7	.867
20% F.1 + 80% G.2	"	12.25	985	195	50	0.98	44.0	32.3	46.5	34.6	"	9.84	50.2	35.9	"	43.2	29.0	28.90	196.1	.886
100% G.2	"	13.81	998	254	70	1.78	35.5	25.8	35.2	25.6	"	11.23	40.8	28.7	Oct.30	36.6	23.2	20.0	201.8	.904
Station Fuel	"	10.59	998	168	39	0.65	64.0	47.3	63.3	49.4	"	7.98	71.2	52.0	Oct.27	61.4	44.4	47.32	186.5	.850
100% High Reference Fuel	"	8.79	968	100	32	0.32		77.0		79.8	"	6.99		74.7	(9)		77.0	73.23	173.2	.806
90% " " "	"	9.23	996	131	32	0.42		69.0		72.0	"	7.17		66.6	"		69.2	61.27	179.7	.828
75% " " "	"	10.02	995	151	33	0.50		57.0		59.4	"	7.42		54.9	"		56.1	43.27	190.2	.862
60% " " "	"	10.69	980	185	39	0.73		45.0		46.2	"	9.04		43.4	"		43.2	29.51	201.3	.894
45% " " "	"	12.18	1005	200	50	1.00		33.2		33.6	"	10.49		31.8	"		30.3	18.05	209.6	.927
35% " " "	"	14.17	989	269	70	1.87		25.3		25.7	"	12.59		24.1	"		22.1	11.32	214.9	.950
Station Fuel	"	10.75	978	162	36	0.58	62.0	46.7	68.0	53.0	"	7.93	72.0	52.6	Oct.27	61.4	44.4	47.32	186.5	.850
100% F.1 + 1% C H NO	"	9.10	986	99	28	0.27	92.6	71.2	110.0	85.5	"	6.96	100.0	74.7	Oct.30	96.8	74.8	70.7	175.7	.813
80% F.1 + 20% G.2 + 1% $C_2H_5NO_3$...	"	9.49	981	119	30	0.35	84.0	64.2	95.0	76.0	"	7.12	89.2	65.9	Nov.2	83.0	63.0	59.0	181.0	.832
60% F.1 + 40% G.2 + 1% $C_2H_5NO_3$...	"	9.78	993	138	32	0.45	78.2	59.5	81.3	65.0	"	7.36	82.8	60.8	"	72.9	54.1	48.0	186.0	.851
40% F.1 + 60% G.2 + 1% $C_2H_5NO_3$...	"	10.33	987	145	34	0.49	68.8	52.2	76.2	60.3	"	8.05	70.1	51.1	Nov.4	63.4	46.2	37.4	191.0	.871
20% F.1 + 80% G.2 + 1% $C_2H_5NO_3$...	"	11.21	971	156	39	0.60	55.5	41.5	67.0	52.2	"	9.05	57.7	41.6	"	55.8	39.5	28.1	196.8	.890
100% G.2 + 1% $C_2H_5NO_3$...	"	11.78	992	189	49	0.92	48.8	36.2	49.0	37.0	"	10.38	46.1	32.6	"	51.9	36.2	19.5	202.0	.908
Station Fuel	"	10.48	982	153	38	0.59	66.0	49.7	67.0	52.3	"	7.96	71.5	52.1	Oct.27	61.4	44.4	47.32	186.5	.850
Station Fuel	Oct.15	10.24	1006	136	37	0.51	67.0	50.6	67.0	52.3	Oct.21	7.72	71.7	52.2	Oct.27	61.4	44.4	47.32	186.5	.850
100% F.1 + 3% $C_2H_5NO_3$	"	6.95	990	79	27	0.21	(12)		(12)		"	6.79	(12)		Nov.11	105.2	82.4	66.9	178.7	.822
40% F.1 + 60% G.2 + 3% $C_2H_5NO_3$...	"	9.08	1028	103	30	0.31	89.0	68.0	92.0	73.8	"	7.71	72.0	52.6	Nov.5	76.0	57.0	35.5	193.1	.879
100% G.2 + 3% $C_2H_5NO_3$...	"	10.19	988	133	36	0.48	68.0	51.4	68.7	54.0	"	9.20	54.8	39.3	Nov.4	63.6	46.2	18.2	204.0	.915
100% F.1 + 5% $C_2H_5NO_3$	"	6.28	972	73	25	0.18	(12)		(12)		"	6.74	(12)		Nov.11	106.6	83.6	63.6	180.9	.830
40% F.1 + 60% G.2 + 5% $C_2H_5NO_3$...	"	8.45	992	97	26	0.26	106.0	82.2	110.0	85.5	"	7.43	77.4	56.6	Nov.5	82.5	62.5	33.5	195.4	.886
100% G.2 + 5% $C_2H_5NO_3$...	"	9.52	986	120	31	0.37	79.5	60.4	81.0	64.5	"	8.85	58.2	42.1	Nov.4	73.8	55.0	17.1	206.4	.922
100% High Reference Fuel	"	8.64	966	92	31	0.28		57.0		79.8	"	6.93		74.7	(9)		77.0	73.23	173.2	.806
90% " " "	"	9.44	966	112	29	0.32		69.0		72.0	"	7.05		66.6	"		69.2	61.27	179.7	.828
75% " " "	"	10.17	1018	133	33	0.43		57.0		59.4	"	7.36		54.9	"		56.1	43.27	190.2	.862
60% " " "	"	10.58	1008	151	40	0.60		45.0		46.2	"	8.67		43.4	"		43.2	29.51	201.3	.894
45% " " "	"	12.07	1015	207	57	1.18		33.2		33.6	"	10.37		31.8	"		30.3	18.05	209.6	.927
35% " " "	"	13.73	1019	300	77	2.32		25.3		25.7	"	12.19		24.1	"		22.1	11.32	214.9	.950
Station Fuel	"	10.17	983	131	35	0.45	68.0	51.4	68.0	53.0	"	7.84	69.9	50.8	Oct.27	61.4	44.4	47.32	186.5	.850
Submarine Engine Secured. Following Data Taken at 675 rpm, 40 BHP and Injector Timing at 1° BTC for Comparison with Report, Serial Number EES-6395-D																				
Station Fuel	Oct.15	10.77	788	129	41	0.53	67.0		67.3		Oct.21	7.84	69.9	50.8	Oct.27	61.4	44.4	47.32	186.5	.850
100% High Reference Fuel	"	8.51	765	84	26	0.22					"	6.93		74.7	(9)		77.0	73.23	173.2	.806
90% " " "	"	9.57	784	88	27	0.23					"	7.05		66.6	"		69.2	61.27	179.7	.828
75% " " "	"	10.22	781	109	35	0.38					"	7.36		54.9	"		56.1	43.27	190.2	.862
60% " " "	"	11.51	766	131	44	0.58					"	8.67		43.4	"		43.2	29.51	201.3	.894
45% " " "	"	13.24	793	240	77	1.85					"	10.37		31.8	"		30.3	18.05	209.6	.927
35% " " "	"	16.40	836	307	104	3.18					"	12.19		24.1	"		22.1	11.32	214.9	.950
Station Fuel	"	10.57	757	113	37	0.41	69.0		70.0		"	7.91	69.1	50.2	Oct.27	61.4	44.4	47.32	186.5	.850
Station Fuel	Nov.3	9.75	960	136	39	0.53	63.5	47.9	70.6	55.5	-	-	-	-	Oct.27	61.4	44.4	47.32	186.5	.850
E.6	"	8.96	973	139	38	0.53	77.5	59.0	70.6	55.5	Nov.19	7.44	70.6	51.1	Nov.10	54.7	38.7	53.89	183.2	.834
Diesel Fuel No.4	"	9.70	1004	156	38	0.59	64.7	48.8	66.7	52.0	"	7.59	68.7	48.6	Nov.11	54.3	38.3	50.06	184.9	.841
" " No.5	"	12.10	1015	232	63	1.46	39.5	29.0	39.4	29.8	"	10.21	43.9	31.0	"	42.1	28.0	39.95	192.7	.869
100% High Reference Fuel	"	8.12	993	92	32	0.29		77.0		79.8	Nov.20	6.50		74.7	(9)		77.0	73.23	173.2	.806
40% " " "	"	8.44	989	102	31	0.32		69.0		72.0	"	6.88		66.6	"		68.2	61.27	180.3	.828
80% " " "	"	8.97	980	127	34	0.43		61.0		63.7	"	7.29		58.8	"		60.4	48.80	187.5	.850
70% " " "	"	9.37	975	143	38	0.54		53.0		54.8	"	7.83		51.0	"		51.8	38.60	194.4	.872
60% " " "	"	10.01	942	154	39	0.60		45.0		46.2	"	8.50		43.4	"		43.2	29.50	201.3	.894
50% " " "	"	10.86	990	205	54	1.10		37.0		37.8	"	9.45		35.7	"		34.6	21.40	206.8	.916
40% " " "	"	12.03	994	228	62	1.41		29.3		29.3	"	10.70		28.0	"		26.2	14.60	212.2	.938
30% " " "	"	14.37	952	328	88	2.88		21.5		21.5	"	13.85		20.5	"		-	8.40	217.4	.959
Station Fuel	"	9.83	994	155	40	0.62	62.5	47.1	65.0	50.7	-	-	-	-	Oct.27	61.4	44.4	47.32	186.5	.850
80% Cetane Primary Fuel	"	8.08	976	102	30	0.30	103.0		101.0		Nov.20	6.31	106.5	-	(9)	102.4		67.9	175.7	.815
70% " " "	"	8.34	1009	128	33	0.42	92.5		86.2		-	-	-	-	"	91.1		53.9	182.8	.841
60% " " "	"	8.94	1015	147	36	0.53	77.5		75.0		Nov.20	7.21	81.5		"	79.6		41.94	189.2	.866
50% " " "	"	9.73	994	155	40	0.62	66.5		64.4		"	7.90	68.8		"	68.1		31.36	199.0	.891
40% " " "	"	10.49	974	176	43	0.76	54.2		52.6		"	8.89	55.4		"	56.5		21.97	208.0	.916
30% " " "	"	12.05	993	224	62	1.39	39.8		40.7		"	10.39	42.5		"	44.5		13.73	214.0	.941
20% " " "	"	14.67	1046	370	102	3.79	29.5		28.7		"	14.35	29.4		"	35.9		6.61	219.3	.967
Station Fuel	"	9.99	991	150	39	0.59	60.5	45.0	66.5	51.6	-	-	-	-	Oct.27	61.4	44.4	47.32	186.5	.850

(7) Average critical compression ratio from thirteen day curves from May 6 to 27, 1936, during which period the C, D, E, and G series fuels were tested for critical compression ratio but not tested consecutively as series fuels. Average reference fuel number from Plate 8 of report, Serial Number EES-6395-C.

(8) Critical compression ratio and reference fuel number on one or more days from May 6 to 27, 1936.

(9) Taken from calibration curve of cetane and reference fuel blends, Plate 8 of this report.

(10) Irregular engine operation caused by residues L. fuel.

(11) Cetane number determined from an average curve of critical compression ratio versus cetane number for all fuels tested for reports, Serial Numbers EES-6395-C and 6395-D. Reference fuel number determined from calibration curve of cetane and reference fuel blends, Plate 8 of this report.

(12) Over 110% reference fuel number.

Fig. 2 (*Continued*)

fore, longer than the true delay by the length of time required for the pressure wave to traverse this gas passage. The two errors are opposite and, from rough calculations, appear to be about equal. Comparison with the work of other investigators would indicate that the recorded delays are between zero and 5 per cent too low. Since, however, the conclusions drawn from these tests were arrived at by comparison of the test fuels with standard primary and secondary reference fuels and since engine operating conditions were maintained carefully during all runs, it has been assumed that the error is substantially the same for all runs and that the value of the comparisons has not been affected adversely.

For ignition-delay testing, the engine was equipped with a special three-tank fuel system. The fuel piping was arranged

for rapid flushing of the lines. Test fuels were filtered into the tanks, and engine filters were bypassed to reduce entrained fuel volume. During preliminary testing lack of consistency between successive engine cycles was traced to pressure waves in the primary fuel system. A wave trap was installed in the injector body, and a bypass line was provided which made it possible to circulate several times through the injector supply chamber the quantity of oil actually taken into the high-pressure pump barrel on the suction stroke. As a result of these modifications, cyclic reproduction within very narrow limits was obtained, and the time required for a complete shift-over from one fuel to another was shortened to 45 sec. However, in normal testing a 20-min. run was made on each sample, the first 14 min. being used to stabilize engine conditions, the next 3 min. for oscillography, and the final 3 min. for taking indicator cards. From 10 to 15 unknowns can be rated in one day. The reference fuel day curve was run in the middle of each day.

In the preliminary work[9] the submarine-type engine was operated at reduced load and speed and with retarded timing to aggravate the effects of ignition delay. In the present work, all running for record was done at rated load and speed and with approved injection setting. These conditions were selected because they are most representative of service requirements, because test procedure had improved to the point where consistent results could be obtained without artificially lengthening the ignition delay, and because the computed combustion knocks and fuel consumptions are more vital under maximum output.

In addition to the indicator diagrams and oscillograms taken for record purposes, a complete log of engine conditions was kept during the entire time the engine was in operation.

During routine testing, four indicator cards and four oscillograms were made at approximate 15-sec. intervals for each test fuel and each reference-fuel blend. At the end of the day's work, cards and oscillograms were measured and averages struck for the four readings in each set. Day curves of ignition-delay angles and computed combustion knock were then plotted from the reference-fuel data, and the reference-fuel numbers of the day's run of test fuels picked from the day curves. These reference-fuel numbers later were transposed to cetane numbers by means of the appropriate calibration curves. Although curves for the secondary reference fuels were run on every test day, only one complete run was made with cetane

[9] See Report 6395-D, May, 1936, Naval Engineering Experiment Station.

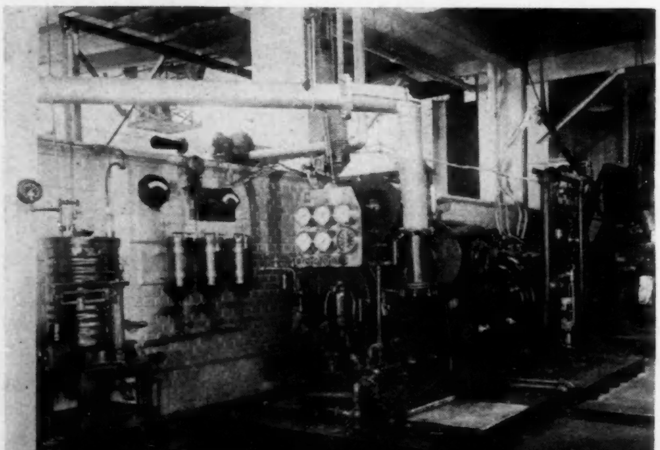


Fig. 3 – Submarine-Type Test Engine and Equipment Used for Full-Scale Testing

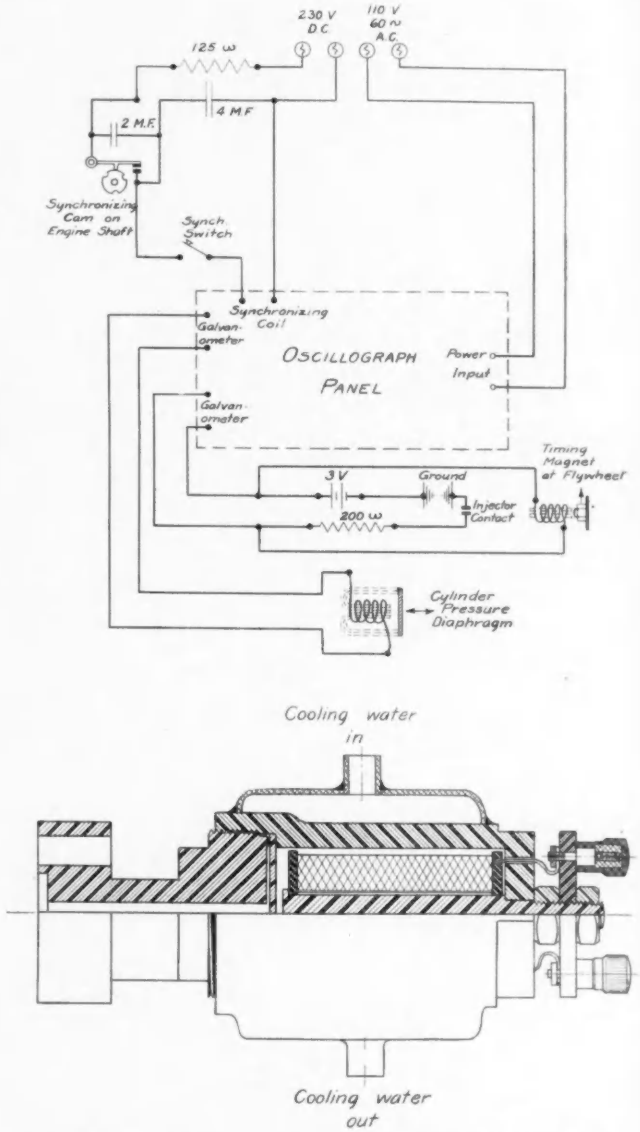


Fig. 4 – Diagram of Oscillograph Circuits and Point-of-Ignition Indicator

due to the scarcity of obtainable supplies and the expense of running a 75 b.hp. engine on $38-per-gal. fuel. The complete test records now include over 1400 indicator cards and an equal number of oscillograms.

The sample indicator cards and oscillograms presented in Fig. 5 were selected and arranged to show as graphically as possible the increases in ignition delay and combustion shock that accompany reduction in fuel ignition quality. Accordingly a high-cetane fuel is compared with a low-cetane fuel, and each reference fuel is roughly equivalent to the cetane blend shown immediately below it. From the indicator cards is apparent the growing engine roughness resulting from increasing explosion-pressure rise and accelerating burning rate, combining to produce high values of the computed combustion knock. From the oscillograms can be seen the lengthening ignition-delay angle, as well as approximate values of peak pressures and rates of pressure rise. At the low end of the ignition-quality range, the ignition-delay angle is nearly equal to the period of injection. In deference to authenticity, these data are presented without photographic retouching. These cards and oscillograms were all made on the same day and under engine conditions controlled as accurately as possible to standard test conditions. In this connection it should be noted that the engine is not only operated under normal governor control, but that the particular fuel-injection system used auto-

matically advances the injection timing with increase in quantity of fuel pumped. Under these conditions, and to a somewhat limited extent, injection timing automatically approaches the optimum advance for the fuel in use.

Summarized results of the engine tests and ignition-quality indexes deduced therefrom or calculated from the physical and chemical properties, appear in Fig. 2. In the case of the primary reference fuels, cetane and alpha-methylnaphthalene, the approved distillation procedure was found inadequate for blends of two substances having such narrow and widely different boiling ranges. For this reason calculated values have been substituted for experimental observations. Even so, the 50 per cent boiling points of the blends are rather indeterminate, and the boiling point-gravity constant dependent thereon may be subject to considerable error.

Curves of calibration of the secondary reference fuels in terms of the primary standards by four different engine methods are given in Fig. 7. The two methods used for the operation of the C.F.R. engine were the most recent prescribed by the Volunteer Group for Compression-Ignition Research. In the knockmeter-delay method difficulty was experienced in rating fuels below 25 to 30 cetane number because of unstable combustion.

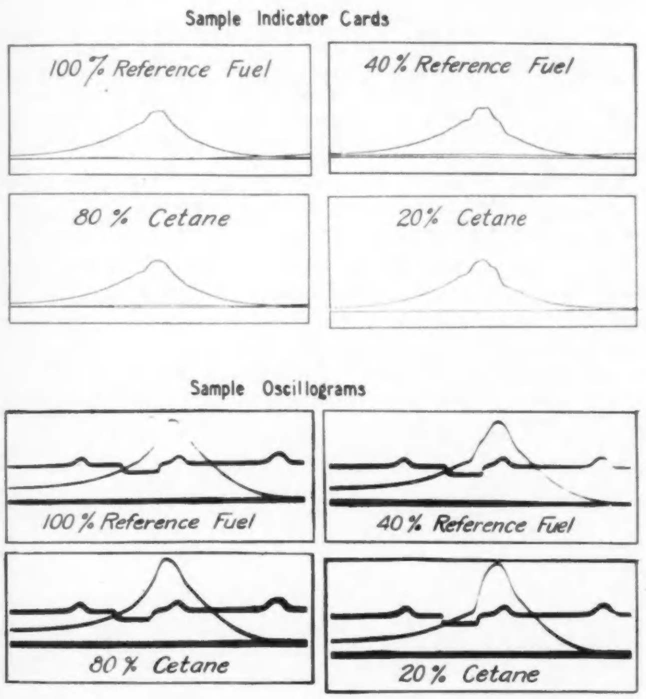


Fig. 5 – Sample Indicator Cards and Oscillograms Showing Progressive Increase in Ignition Delay and Combustion Knock with Decreasing Ignition Quality

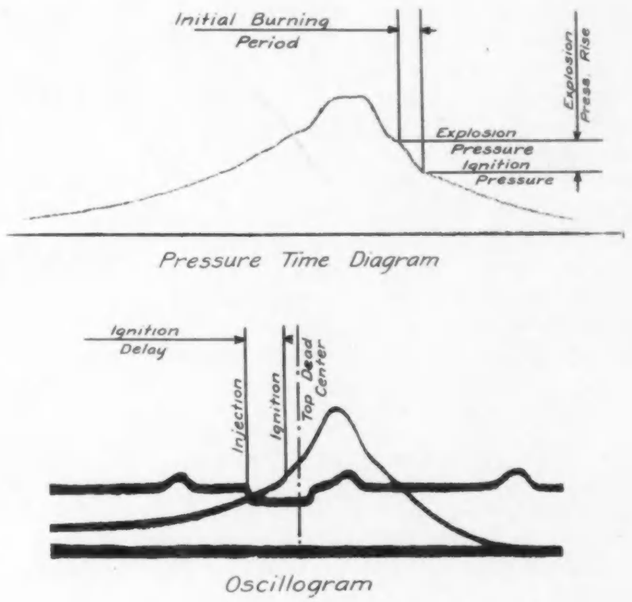


Fig. 6 – Method of Measuring Indicator Cards and Oscillograms

Discussion

Discussion of the tests reported in this paper is predicated on the writer's conviction that ignition quality, as applied to Diesel fuel, is a property inherent in the fuel itself, as constant for any given fuel as the other recognized properties, although tempered by the concession that ignition quality is related to and dependent upon many of these other properties. Ignition quality, by this conception, beggars definition and is measurable only through its effects. It should be measured, therefore, in the place where those effects are most vital, that is, in full-scale engines in field service.

Ignition delay, defined as the interval in time or in degrees of crankshaft rotation elapsing between the start of injection and the commencement of discernible combustion as determined by the pressure rise from the compression line and generally considered to be the fundamental manifestation of ignition quality in the running engine, is not a characteristic of the fuel alone but a property of the entire system, embracing not only the fuel itself, but the engine in which the fuel is burned, and the conditions under which the engine is operated. Ignition delay can, therefore, be considered as only one of several factors which, taken together, delimit the effectiveness of fuel utilization.

The consumer of Diesel fuels is concerned with effects – primarily with smooth combustion and low specific fuel consumption simply because reduction in repair and fuel bills come respectively with smooth and efficient combustion. As an arbitrary yardstick by which to measure smoothness of combustion,

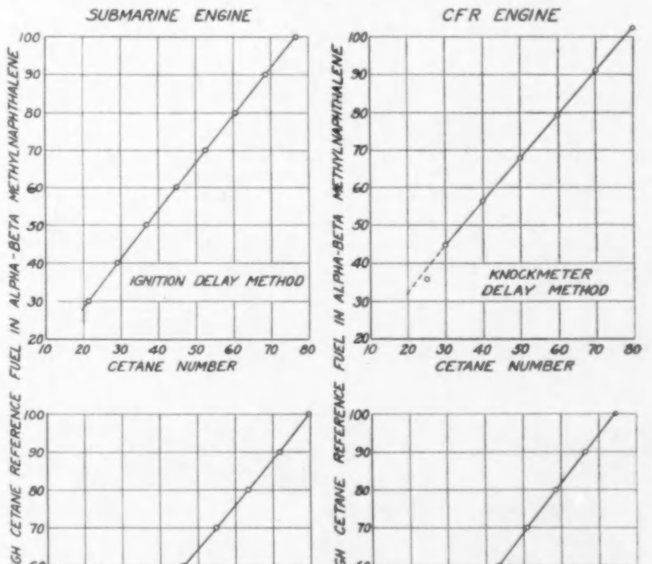


Fig. 7 – Calibration Curves of Secondary Reference Fuels

the computed combustion knock,[10] first proposed by Joachim, has much to recommend it. It possesses for the operating engineer a tangible significance; it can be heard and felt in the running machine. It is possible of measurement in the field with the ordinary instruments usually available.

The definition of computed combustion knock is built up from two subordinate definitions: the explosion pressure rise, described as the first fast increase in pressure resulting from the rapid combustion of that part of the fuel charge deposited in the engine cylinder during the ignition-delay period; and the burning rate, defined as the explosion pressure rise divided by the interval, expressed in crank-angle degrees during which this phenomenon occurs. Computed combustion knock is the product of explosion pressure rise and burning rate divided by 10 to the fourth power:

$$C.K. = \frac{E.P.R. \times B.R.}{10^4}$$

The arbitrary number thus obtained is not in itself significant. It is as a basis for comparisons between fuels that it acquires value, for it shows relatively the *effects of the compared fuels on the service engine.* Its utility becomes paramount when used to rate commercial fuels in terms of the standard reference fuels, for from this practice comes the life-size reference fuel or cetane number of real meaning to the operator.

With these considerations as a background, the remainder of the test results are discussed in sequence.

Fig. 8 is a plot of the fundamental data obtained from the full-scale tests: ignition delay in crank-angle degrees vs. the computed combustion knock. Although the spread of points is considerable, it should be noted that this is basic data without correction for day-to-day effects, and that the spread is increased by the inclusion of the doped fuels. This curve is semi-logarithmic in character.

In Fig. 9, the basic data of the preceding chart have been transformed into cetane numbers by conversion from ignition delay and computed knock to reference-fuel numbers through the day curves, and from this point to cetane numbers by the calibration curves of Fig. 7. The curve presented is the mean of all the fuels, doped, undoped, and reference. The preponderance of undoped-fuel data points is above the line; the preponderance of doped fuel data below; whereas the reference blends fall on the average line within the limits of experimental accuracy. For the cetane numbers themselves, in the range from 20 to 50 cetane, both methods give approximately equal results. In the range of high-quality fuels, from 50 to 80 cetane, the computed knock method rates the fuels progressively higher, until an 80 cetane by this method is equivalent to a 75 by the ignition-delay method.

The correlation between the submarine-engine, computed knock method, and the two C.F.R. engine determinations is shown in Figs. 10 and 11. In Fig. 10, that for the C.F.R. critical-compression-ratio method (upper right-hand corner), it is found that the correlation fails for the doped fuels. The heavy line represents the average of the undoped fuels and shows that, for this class of fuels, determinations by this method are satisfactory, with a life-size cetane number of 80 corresponding to a C.F.R.-critical-compression-ratio number of 73. The addition of ethyl nitrate causes an increase in the submarine-engine cetane number without a corresponding decrease in critical compression ratio, indicating that, although combustion may be improved by this dope, starting will not be facilitated equally. The secondary doping with tetraethyl lead reduces the cetane number without appreciable effect on the critical compression ratio. The correlation with the C.F.R. knockmeter-delay data (Fig. 11) is much better. Again the curve represents the mean of all fuels, with the weight of undoped fuels above and of the doped fuels below the line. At the ends of the normal range of ignition quality, submarine-engine computed knock cetane numbers of 25 and 80 correspond to C.F.R. knockmeter-delay cetane numbers of 24.5 and 70 respectively.

Fig. 12, showing the cetane numbers by submarine-engine ignition-delay vs. the C.F.R. knockmeter-delay cetane numbers, completes the series of engine comparisons. This curve likewise represents the average of all fuels, but the separation of undoped fuels above the line and doped fuels below is less pronounced. Again, at the low end of the ignition-quality range, the cetane numbers by the two methods are approximately the same whereas, at the upper end, the submarine rating of 80 corresponds to a C.F.R. rating of 76.5.

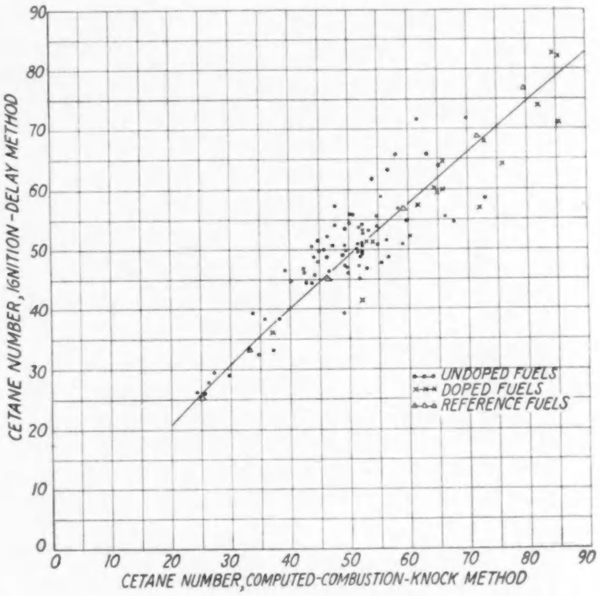


Fig. 9 – Cetane Number, Submarine-Engine Ignition-Delay Method Vs. Cetane Number, Submarine-Engine, Computed-Combustion-Knock Method

[10] See *Diesel Power*, May, 1935, pp. 283-290; "The Characteristics of Diesel Fuel Oil," by W. F. Joachim, presented at the May, 1935, National Meeting, Oil and Gas Power Division of the A.S.M.E., Tulsa, Okla.

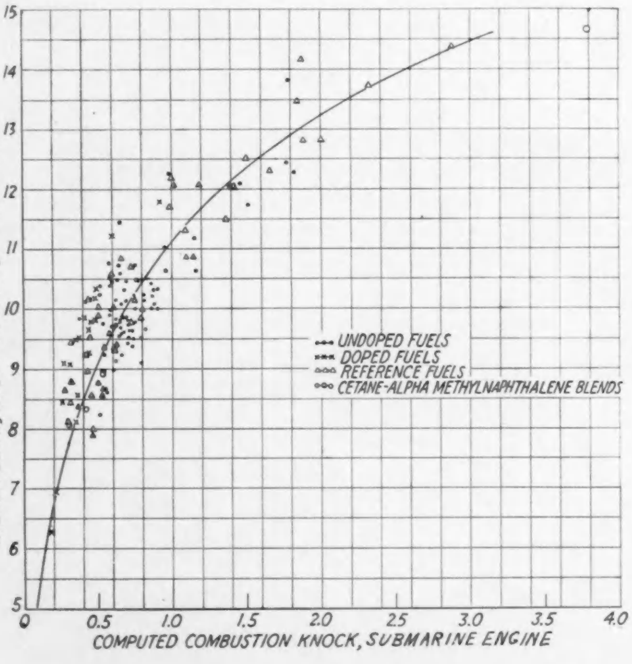


Fig. 8 – Submarine-Engine Ignition Delay Vs. Computed Combustion Knock

When the comparison is shifted from engine indexes to physical-chemical indexes, the picture changes. Figs. 10 and 13 represent, respectively, the correlation between Diesel index, viscosity-gravity constant, and boiling point-gravity constant and the submarine computed knock cetane numbers. In each of these charts the heavy, full-line curve represents the average of undoped fuels, with the spread of data indicated by the location of the individual datum points. The light, full-line curves are for the primary and secondary reference-fuel blends, the dashed lines for the doped series of *F*-1/*G*-2 cross-blends, whereas the dotted lines carry fuel *E*-2 through progressive additions of ethyl nitrate and tetraethyl lead. For the undoped fuels, correlation between physical-chemical indexes and engine performance is only fair; for the doped fuels it fails completely, large increases in cetane numbers being accompanied by slight or reverse changes in the physical-chemical criteria. In passing, mention is made of the mechanics of obtaining cetane ratings over 100. These are approximations, based on extrapolated cross-plots of basic data. In some cases, the numerous cross-plots proved of equal interest to the final results, but space and time prohibited including them in the paper.

To permit rapid inter-comparison the results of all seven of the performed methods of evaluating the ignition quality of Diesel fuels are brought together in Fig. 10, the cetane number obtained by computed combustion knock from full-scale engine performance being used as the base line. It is immediately

DIESEL INDEX; VISCOSITY-GRAVITY CONSTANT; BOILING POINT-GRAVITY CONSTANT; CETANE NUMBERS, CFR ENGINE, KNOCKMETER DELAY AND CRITICAL COMPRESSION RATIO METHODS, AND CETANE NUMBER, SUBMARINE ENGINE, IGNITION DELAY METHOD VS CETANE NUMBER, SUBMARINE ENGINE, COMPUTED COMBUSTION KNOCK METHOD

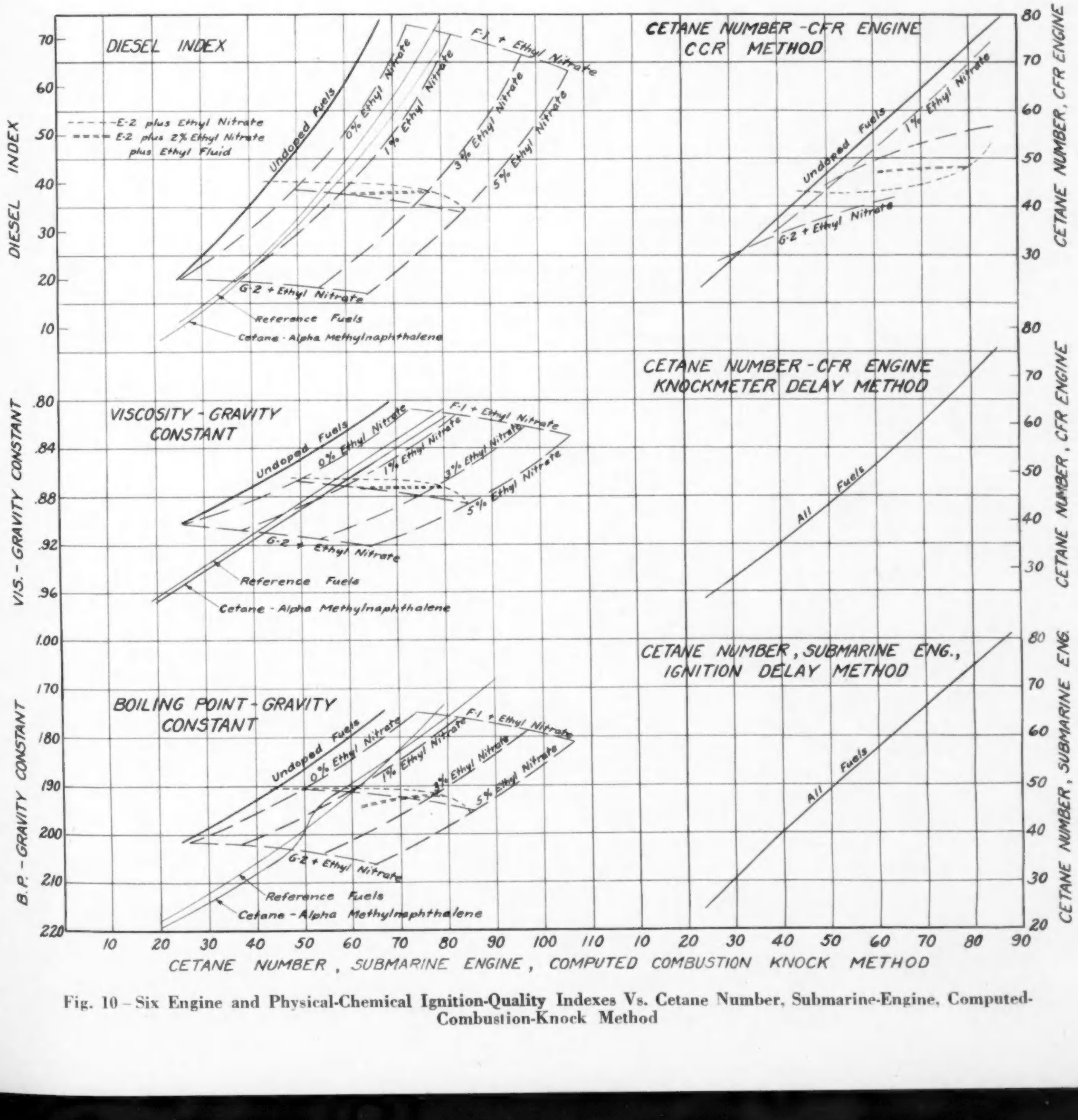


Fig. 10 – Six Engine and Physical-Chemical Ignition-Quality Indexes Vs. Cetane Number, Submarine-Engine, Computed-Combustion-Knock Method

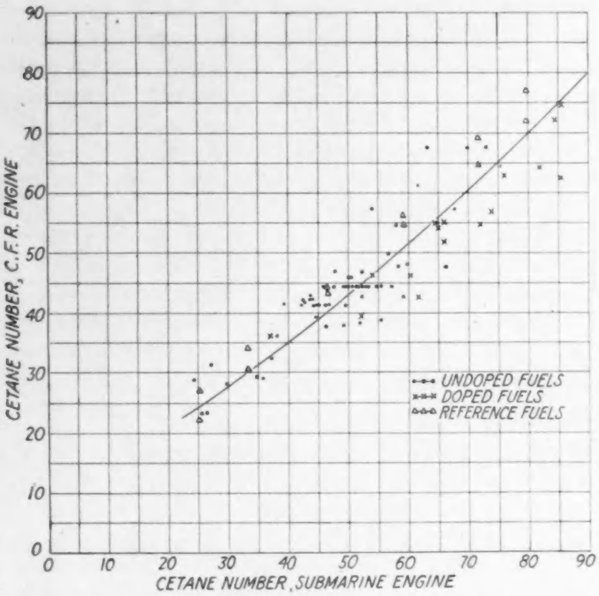


Fig. 11 – Cetane Number, C.F.R. Engine Knockmeter-Delay Method Vs. Cetane Number, Submarine-Engine, Computed-Combustion-Knock Method

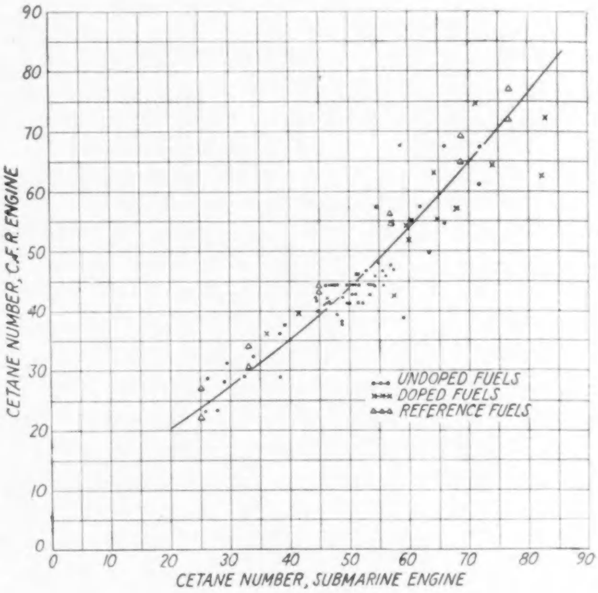


Fig. 12 – Cetane Number, C.F.R. Engine Knockmeter-Delay Method Vs. Cetane Number, Submarine Engine, Ignition-Delay Method

apparent that, of these seven methods, only the C.F.R. engine, knockmeter-delay, and the submarine-engine ignition-delay and computed-combustion-knock methods afford satisfactory predictions of service performance for all fuels.

Analysis of all the data accumulated from these tests has not been completed. Sufficient evidence has been obtained to indicate that comparisons between reference and test fuels, recorded in cetane numbers, may be obtained through examination of other important combustion phenomena, such as maximum cylinder pressures, explosion-pressure rises, burning rates, and fuel consumptions, although the difficulties in the last-mentioned item are aggravated by the wide difference in heat content between cetane and alpha-methylnaphthalene. Likewise, it is probable that other of the several engine tests or methods of instrumentation now under investigation in laboratories throughout the country will develop equal or better correlations. The efficacy of such methods, in general, may be judged by how well they compare with the C.F.R.-knockmeter-delay determinations. Should it be found practicable to re-instrument the C.F.R. engine so that it measures not ignition-delay but the dynamic effect of combustion shock directly, such a method should prove the most satisfactory of all.

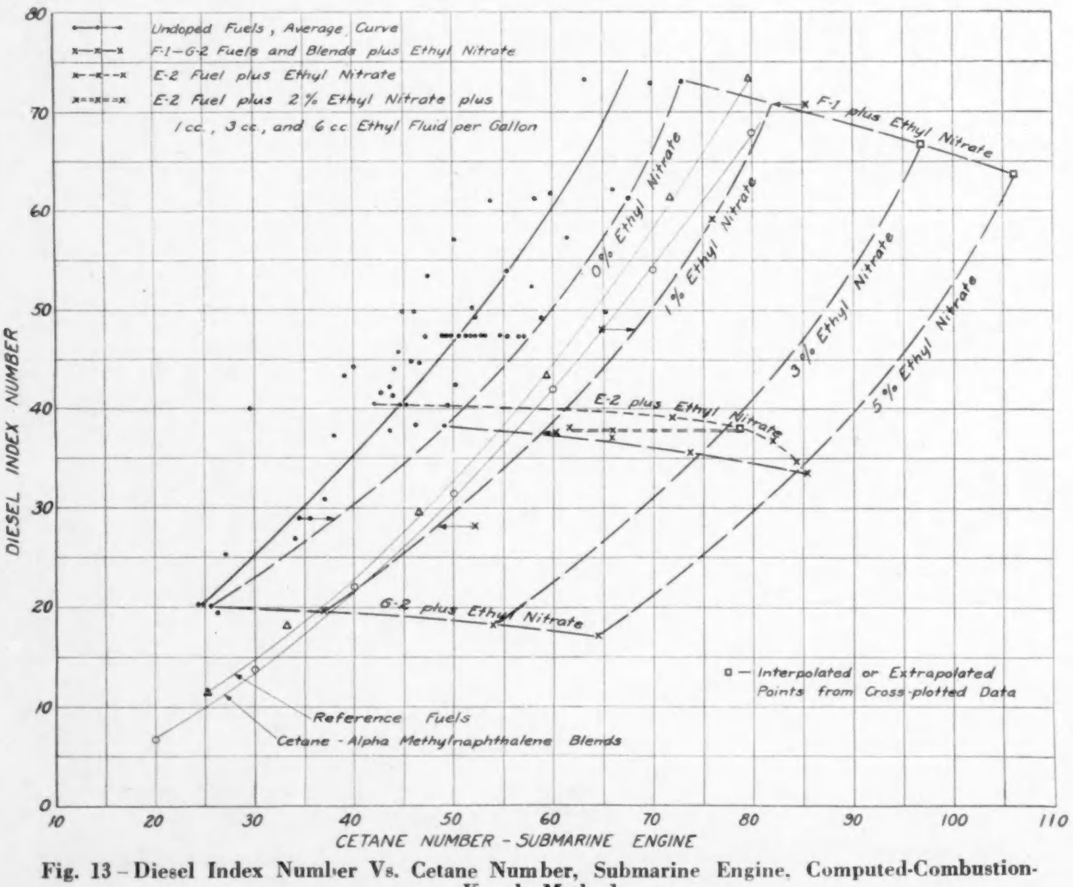


Fig. 13 – Diesel Index Number Vs. Cetane Number, Submarine Engine, Computed-Combustion-Knock Method

Conclusions

The evaluation of the ignition quality of Diesel fuels in service engines in field use by comparison with accepted standard reference fuels is possible. Ordinary instruments and procedures are sufficient. Results are of practical significance and reproducible within limits necessary for safe application. Either ignition delay or computed combustion knock may be used as the basis for comparison, depending on which measurement is obtained more easily under the existing circumstances.

The cetane numbers of Diesel fuels obtained in the laboratory using the knockmeter-delay method in the Cooperative Fuel Research engine provide a reliable forecast of those oper-

ating characteristics of that class of engines, represented by the submarine test unit, which depend on the ignition quality of the Diesel fuel used, regardless of whether the fuel is doped or is an unadulterated petroleum derivative. In general, the cetane numbers of low-ignition-quality fuels are equal when determined in the laboratory or by either method in the service engine. Medium- and high-ignition-quality fuels are rated progressively higher by the latter engine. At the upper end of the normal range of ignition quality, cetane number 70 by the Cooperative Fuel Research engine, knockmeter-delay method compares with cetane numbers of 74.5 and 80 in the submarine engine by ignition-delay and computed-combustion-knock methods, respectively. The laboratory rating is conservative; it may be used with assurance.

The laboratory method of Cooperative Fuel Research engine determinations using the critical compression ratio as the basis for comparison fails to correlate with service-engine determinations for abnormal fuels. As a corollary, the use of ethyl nitrate improves combustion characteristics in the service engine, but shows less promise of equal improvement in startability.

The ignition-quality indexes derived from physical and chemical examination of fuels show fair correlation with service performance for normally refined fuels, but are unreliable in evaluating the ignition quality of doped fuels.

The early adoption of a standard laboratory method of operating and instrumenting the Cooperative Fuel Research engine, and vigorous pursuit of correlating tests in field engines of representative combustion characteristics and sizes is highly desirable. If the Cooperative Fuel Research engine can be reinstrumented to compare fuels on the basis of dynamic combustion shock, this work will be facilitated and the degree of correlation obtainable will be improved.

Discussion

Comment on Indicator-Card Forms Requested

– Charles S. Moore

Assistant Mechanical Engineer, National Advisory Committee for Aeronautics

I WISH to submit for comment by Lt.-Commander Good some indicator-card forms which are different from those presented in his paper. In these cards the entire pressure rise, if any, is quite a smooth curve and does not contain the well-defined initial burning period measured by Lt.-Commander Good unless the whole pressure rise is considered. Each of these cards was obtained using the same fuel and same test engine and equipment. Engine operating conditions were different for each card and determined the card form.

Fig. *A* is an indicator card from a 5-in. by 7-in. single-cylinder test engine at 2000 r.p.m.; it is shown in Fig. 18(*a*) of "Compression-Ignition Engine Performance at Altitude," by the author and John H. Collins, Jr., p. 272, this issue. Altitude conditions are at 14,000 ft. (moderate knock). The pressure-rise line is practically a straight line.

Fig. *B* shows an indicator card for inlet-air temperature of 258 deg. fahr. (very smooth operation); it is shown in Fig. 18(*d*) of "Compression-Ignition Engine Performance at Altitude," p. 272, this issue. No well-defined initial pressure rise is apparent unless the whole pressure rise is taken.

Fig. *C* is illustrated opposite.

When this card was taken operation was very smooth as computed combustion knock would indicate, but how can fuel be rated from such a card? This constant-pressure combustion probably can be attained from a number of different fuels, even at high rotational speeds because, in the card presented, compression and injection conditions controlled the card shape.

In conclusion then, will it not be necessary to test all fuels under identical conditions in order to rate them properly?

Believes Additional Factors Influence Ignition Quality

– Nicholas P. Setchkin

Consulting Engineer

LT.-COMMANDER GOOD made an excellent step forward in his attempt to bring before all a correlation among data secured by different methods recommended for qualification of Diesel fuels and a comparative analysis of the relationship obtained in evaluating consistency and significance of one or other recommended methods of rating of Diesel fuels for actual service.

In starting this comparative analysis, Lt.-Commander Good made the conditional definition of his conception of "quality of Diesel fuel," which is stated as follows: "Ignition quality is a property inherent in *fuel itself*, as *constant* for any given fuel as the other recognized properties."

Further on he defines his other conception of the "ignition delay," which in his opinion *is not* the "ignition quality" because it is related to the running engine, therefore, is not a characteristic of *fuel alone*, but a property of the *"entire system in operation."*

Still further on Lt.-Commander Good gives thorough determination of "arbitrary yardstick of smoothness of combustion or the dynamic characteristics of a combustion phenomena" or as he called the "computed combustion knock" which is a product of the "explosion-pressure rise" times the "burning rate." Of course, neither "explosion-pressure rise" nor "burning rate" are the "properties inherent in fuel itself," therefore, this type of characteristic as well as the "ignition lag" are not the "ignition quality" in the sense of his stated conception and, therefore, they should not be used for evaluation of Diesel fuels.

It would be interesting to clarify this discrepancy between this conception of the "ignition quality" and the recommendation to qualify fuels by reference fuels, which are interrelated by such variable properties as are the "ignition lag," "combustion knock," critical compression ratio, and so on.

The study of correlation of all obtained data plotted shows that practically all points are concentrated on one side of equilibrium (45 deg.) line, and deviation is gradually increasing from the low-cetane fuels (long ignition lag) to the high-cetane fuels (short lag). However, a separate study of a particular group of fuels, for example, fuels of type *C*, *G*, and blended fuels *F* and *G*, shows that reference fuels and some types of Diesel fuels (type *G*) are consistently close to the equilibrium line. However, other groups such as *C* or *F* plus *G* consistently deviate from it. This finding clearly indicates that selected ignition qualities do not sufficiently cover the subject of qualification and certain important factors are omitted from consideration.

One of these factors may be ignition advance angle, which is used constant for all fuels. Smoothness of operation or proposed "combustion-knock factor" is dependent greatly upon the ignition setting, slight modification of which immediately produces smooth operation on fuel first considered as rough fuel. The similar consideration can be applied to other important factors of engine operation such as the temperature of jacket water, air charge, and many other characteristics. Careful study of these factors is very essential for determination of the "ignition quality" of fuel inherent in the fuel itself.

In reference to Item No. 1 in the fundamental requirements of fuel for Naval Service, the following question is asked: "Would it be practical

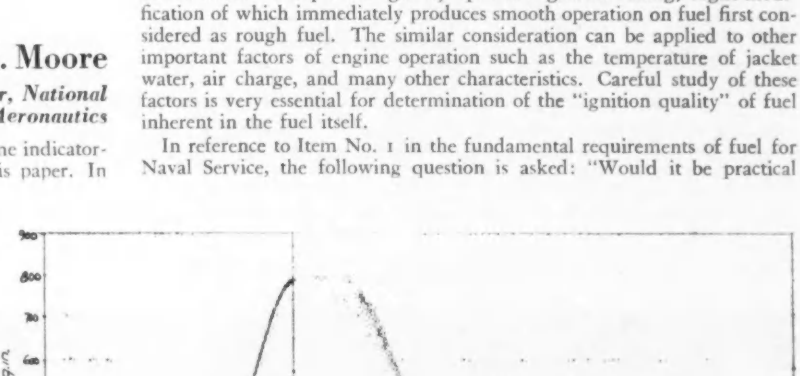


Fig. *C* – (Moore Discussion) Indicator Card for 7½ In. Hg. Boost Pressure and Proper Injection Characteristics – Card Shows Nearly a Flat Top

from the viewpoint of National emergency to operate all Diesel engines only with one grade of fuel suitable for all purposes"?

Would it cause any confusion during certain war conditions if, instead of standard Naval fuel, only ordinary commercial grades were available?

Would it be more useful to classify fuels in different groups and to adopt minor modification of engine adjustment to suit the particular grade of fuel? For example, to specify a slight change of injection setting, or temperature of jacket water, and so on, will open possibilities of efficient use of a much wider range of fuels than to be restricted to one single grade which may not be widely available?

Analysis of great number of accumulated test data, as is stated by Lt.-Commander Good, is not completed, therefore, it will be very interesting to know further findings and conclusions on this subject especially in view of the opinion stated in the author's introduction and concerning his consideration of "every defined property of petroleum derivatives, selection of which has major significance in respect to the fundamental requirement for a good fuel."

This broad statement creates hope that, in further Diesel fuel investigation, many other important factors will be considered for purpose of proper correlation of all essential and indicative properties which define the quality of Diesel fuel.

Illustrates Greater Strains of Operating on Low-Cetane Fuels

– R. Stansfield

Chief Research Engineer, Sunbury Laboratory, Anglo-Iranian Oil Co., Ltd.

LT.-COMMANDER GOOD has given some very useful facts regarding the rating of Diesel fuels in life-size engines. Perhaps other facts about the effect of low-cetane-number fuels on life-size engines also may be of interest.

Fig. *D* is an oscillogram taken from a five-cylinder railcar type of Diesel engine running at a speed of 820 r.p.m. and at about three-quarters full load. The diagram shows the vertical strains in the crankcase structure in line with No. 1 cylinder for the two revolutions of the cycle. It will be noticed that there are several pronounced peaks on this diagram, and a careful study shows that one such peak occurs each time a cylinder fires and that the effect on the part measured depends on the proximity of the firing cylinder to the measured length. The effect of firing in Nos. 2, 3, and 4 cylinders can be seen readily, whereas that of No. 5 cylinder is just discernible.

Such a diagram has a height at any point proportional to the stress in the crankcase metal at that moment in the cycle.

It is common experience among users of Diesel engines that low-cetane fuels are more liable to lead to engine breakdown than are high-cetane fuels, even though a modern pressure indicator may show that the actual peak pressures in the cylinder are not affected much by the fuel used. Fig. *E* taken from the same engine as Fig. *D* but with a low-cetane-number fuel in use shows that the strains are much greater on account of the additional shock or impact loading imparted to the structure by the long-delay fuel, and thus the stresses also are much increased.

How much the impact will increase the stress beyond that determined by calculation from a pressure diagram depends on the precise circumstances but, in the case given, application of the expression given by Lt.-Commander Good gives relative combustion knocks agreeing closely with the relative measured strains.

The diagrams shown are of crankcase strains and it also is easy to prove that the impact from a long-delay fuel has a bad stressing effect on such vital items as the connecting-rod, the crankshaft, and other working parts.

T. B. Rendel has commented on the upper limit of cetane requirement for certain engines. Although it may be true that there are engines which run better on such de-graded fuels it is the writer's experience that such engines are of a less efficient type, and that, the more efficient thermally a Diesel engine is made, the more it depends on a high-cetane-number fuel for best performance.

Contends Ignition Delay Can Be Shortened Too Much

– T. B. Rendel

Shell Petroleum Corp.

MR. POPE has speculated on the question of what might happen should the fuel industry be able to supply a fuel of such ignition quality that the delay would be reduced practically to zero. In many cases this fuel would result in a distinct loss of power and efficiency. Many engines now are of such design that the cetane-number requirement is a maximum rather than a minimum, and if this requirement is exceeded, losses in power up to 15 per cent are experienced. In such cases, if the delay is too short, the fuel is not injected sufficiently far into the combustion-chamber to insure thorough mixing. To carry this point to its extreme implication, one might consider zero delay. The fuel will then burn immediately as it comes from the nozzle, and will not be injected sufficiently far into the combustion-chamber to find the air necessary for combustion in the short space of time available.

I also should like to remark in connection with this question of cetane numbers, that the Volunteer Group for Compression-Ignition Fuel Research is working actively on the question of standardizing measurement of laboratory cetane numbers, and hopes to produce a report recommending a standard procedure for the industry at the next Annual Meeting of the Society. At present we have what we consider a suitable engine and a suitable method, but we have not yet developed a completely satisfactory method of instrumentation for measuring ignition delay.

Improvements Suggested in Instrumentation

– P. H. Schweitzer

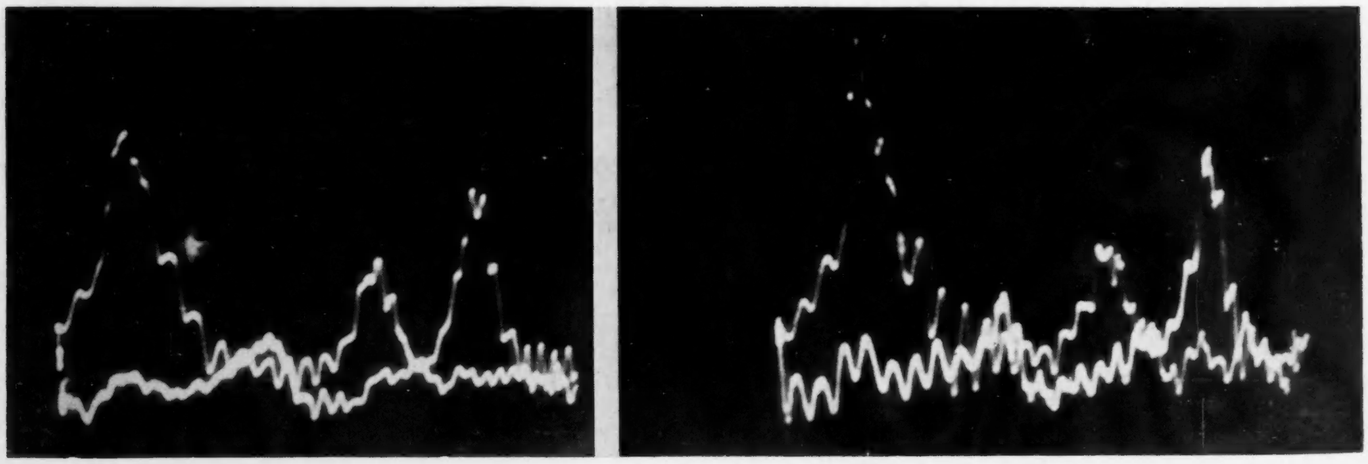
Professor of Engineering Research, The Pennsylvania State College

REGARDING the Navy Diesel-fuel specification, it is the writer's opinion that the three requirements: 0 deg. fahr. pour point, high cetane rating, and production at non-premium price are not easy to reconcile because many crudes that have high cetane rating also have high pour point. The pour point can, of course, always be lowered by de-waxing but not without extra cost.

The author's test of normal and doped fuels in a submarine-type engine did not produce surprises but brought welcome confirmation to theories held by some of us and rejected by others. The most interesting result is that the life-size engine is, if anything, more responsive to fuel ignition quality than is the laboratory engine.

Referring to the oscillograms on Fig. 5, the writer found it more difficult to locate the point of ignition on the oscillograms than on the indicator cards. What is the advantage of an oscillograph with a pickup over a good indicator if it does not make the ignition point more pronounced? Is a pickup which responds purely to the rate of rise not preferable?

(*Continued on page* 251)



Figs. *D* and *E* – (Stansfield Discussion) Oscillograms Taken From a Five-Cylinder Railcar Type Diesel Engine, 820 R.P.M., Three-Fourths Load – Fig. *D* – (*left*) Using High-Cetane Fuel – Fig. *E* – (*right*) Using Low-Cetane Fuel

The Influence of Humidity on Knock Ratings

By J. R. MacGregor
Standard Oil Co. of Calif.

IT has been assumed generally that variations in humidity would not cause errors in knock rating when using the bracketing method. However, the preliminary test results presented indicate that this assumption is not valid for all fuels. Differences in knock ratings of over three octane numbers were found with certain combinations of test and reference fuels when the humidity was varied over the range normally experienced in knock testing.

It was found that the influence of humidity on detonation is not primarily the result of changes in dry air pressure or oxygen concentration, but apparently depends on the nature of the fuel itself.

The results presented are known to be affected somewhat by changes in engine adjustment and bouncing-pin setting. However, for certain fuels the error introduced by humidity changes is considerably greater than the normal experimental error with controlled humidity. Although for some conventional-type fuels humidity control is not required, it is required when testing certain special fuels, leaded gasolines, and octane-heptane blends.

A description is included of the construction of an engine hygrometer and a humidifier both of which have been found convenient to use in routine testing.

THE knock rating of gasolines has undergone a fairly intensive investigation for a period of years. During this time it has been apparent that variables other than those introduced through personal error on the part of the operator were having an important influence on the values obtained. It has been realized that some of these variables are inherent in any test using an engine where cycles are repeated a number of times per minute. Investigations have been made in an attempt to establish the cause of these variations attributable to mechanical causes, and suggestions have been made for controlling them. This paper calls attention to another variable that must be brought under control before consistent, reproducible results can be expected.

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 11, 1937.]

It has long been known that humidity directly affects the amount of detonation in a spark-ignition engine with any fuel, it being much more pronounced at low humidities. In order to overcome this known influence of humidity, as well as to diminish the influence of gradual changes in the mechanical perfection of the test motor, a system of fuel rating has been established which depends upon the bracketing of the unknown fuel between two blends of reference fuels differing but slightly in their knocking tendencies. It has been assumed that, by this means, variations in atmospheric conditions would have no effect on the ratings obtained, and this hypothesis is correct if both reference and test fuels are of the same type.

With the best available control of all known variables and using the bracketing system of reference fuels, variations in results on the same fuel, but tested in different laboratories, or even in the same laboratory at different times, continue to attract attention. In general, these differences have been most pronounced when rating fuels of widely different characteristics, and indicated the influence of some atmospheric variable on the combustion of the fuel. This result suggested that variations in the water vapor present in atmospheric air might be acting differently with different fuels, and an investigation was made to determine the range of atmospheric humidity prevailing in various laboratories throughout the United States for which data were available.

Humidity Range in the United States

During most of 1935 and the latter portion of 1936, a group of approximately 20 laboratories, situated in various parts of the United States, submitted data which included the humidities existing in their laboratories on one day in every month. These data are shown in Fig. 1, and the wide range of humidities indicate that, if humidity has any effect on the relative ratings of gasolines, many of the inconsistencies previously encountered might be explained on this basis. It should be noted that the values shown are based on single monthly determinations and daily, or even hourly, variations might increase appreciably the spread over those indicated. In addition to the maximum, minimum, and average values for both 1935 and that portion of 1936 for which data were available, a curve also is shown which represents the humidities noted in the laboratory of the Standard Oil Co. of Calif. at five-day

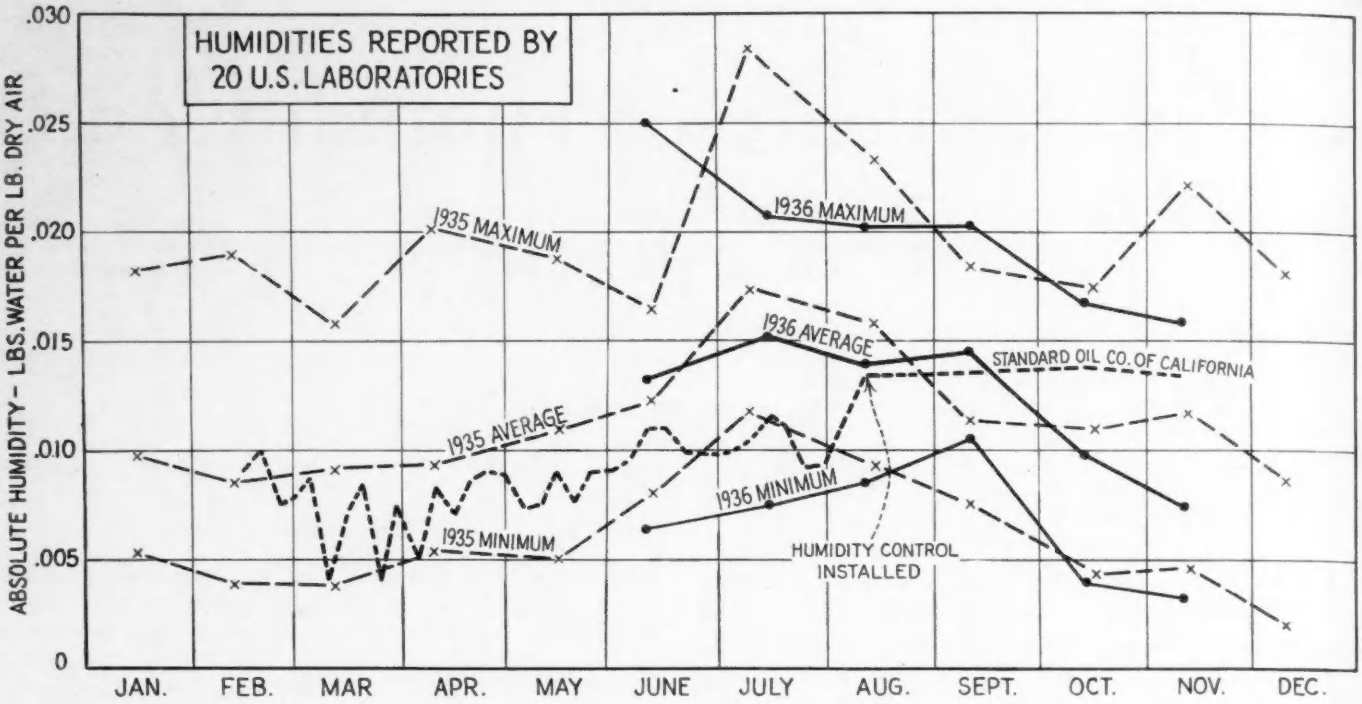


Fig. 1

intervals up to the time humidity control was installed. Since the humidities noted in this laboratory, in general, were lower than the average and far below the maximum values reported, a series of exploratory tests were undertaken to determine the influence of humidity on ratings of gasolines.

Conventional Methods of Measurement

As the first step in an investigation of this nature, it was necessary to obtain a system of laboratory instrumentation and control that would permit the variation in the variable under consideration. Humidity may be expressed as per cent relative humidity, or as absolute humidity in terms of pounds or grains of water per pound of dry air, or in terms of vapor pressure of water in suitable pressure units. The relative humidity values, when reported as such, require a conversion to absolute humidity by use of a corresponding dry-bulb temperature in order to provide a proper basis for comparison, inasmuch as relative humidity is only a ratio and does not indicate a fixed amount of water vapor.

Although the relative humidity is at times as high as 95 per cent at Richmond, Calif., the water-vapor content of the air rarely exceeds 0.010 lb. because of the low atmospheric temperature. On the East Coast, double this value is common with relative humidities of 80 per cent or less. The humidity at the Richmond laboratory rarely falls below 0.005 lb., yet Eastern humidities in the winter fall as low as 0.001 lb. with relative humidities as great as 80 per cent. Relative humidity values, therefore, are not only meaningless but are actually misleading.

It is believed that pounds of water per pound of dry air is the most convenient and significant unit to use, and all values in this paper, therefore, are based on this unit. Vapor pressure of the water in inches of mercury may be converted to equivalent pounds of water per pound of dry air, with negligible error in the normal range, by multiplying by the constant 0.021. A very useful chart for obtaining psychrometric data from the wet- and dry-bulb temperatures appeared in the November, 1935, issue of *Refrigeration Engineering,* p. 276, and is based on the latest thermodynamic data compiled by the General Electric Co. For use in our laboratory an additional scale has been placed at the left of the chart referred to, in order to obviate the necessity of converting from weight in grains to weight in pounds.

Humidity determinations frequently are obtained by means of a sling psychrometer. In using this conventional instrument, however, certain precautions should be observed. The type of material used in the wick, its fit around the thermometer bulb, and the means employed to supply moisture thereto, are of paramount importance. Usually instruments of this nature can be rendered quite suitable, even in the case of relatively poor wicks, by furnishing a small cup of water from which the wick obtains its moisture supply. The old-fashioned hygrometer, consisting of a wet- and dry-bulb thermometer mounted in some form of simple, stationary frame is generally valueless for laboratory work due to the usually stagnant air surrounding it. In this connection attention should be given to local conditions which sometimes give rise to abnormal humidities. For instance, at one time in the laboratory a plugged drain in a nearby trench carrying away jacket cooling water allowed water to accumulate until an exhaust line, located near the top of the trench, was immersed. An abnormally high humidity of double the normal maximum resulted.

Engine Hygrometer

The danger of obtaining erroneous humidity determinations when using conventional measuring methods led to the development of a special engine hygrometer for indicating the humidity of the air actually entering the engine. The unit shown in Fig. 2 has been used with marked success. The dry-bulb thermometer installation is conventional in all respects and requires no further discussion. The wet-bulb thermometer unit, however, is the outcome of a considerable amount of development, and the design shown is quite satisfactory. It will be noted that the wick surrounding the thermometer bulb is moistened continuously by water supplied from a converted drip-feed oiler. The water is conducted from the discharge of this supply through a small tube projecting into the air stream above, and displaced slightly from, the thermometer bulb. The projecting portion of this tube has several small holes drilled radially through it to permit the

escape of water at various points. Surrounding this tube is one end of the wick, the other end of which encloses the thermometer bulb in the conventional manner. The water supply has been provided for convenience, whereas the arrangement shown for wetting the wick is the result of several experimental steps. By this means the water reaching the thermometer-bulb envelope is first cooled to the temperature indicated by the wet bulb before actually contacting that instrument. Without this precaution the water, which is normally at a higher temperature than is the wet bulb, has an undesirable influence on the thermometer indication.

The design shown in Fig. 2 indicates a slip joint between the wet-bulb thermometer holder and the air-intake pipe. This precaution was included originally since it was felt that the moisture evaporated from the wet-bulb wick would add an uncontrolled amount of humidity to the air supplied the engine. In order to evaluate this effect, as well as to eliminate any uncertainties as to the accuracy of the values indicated by

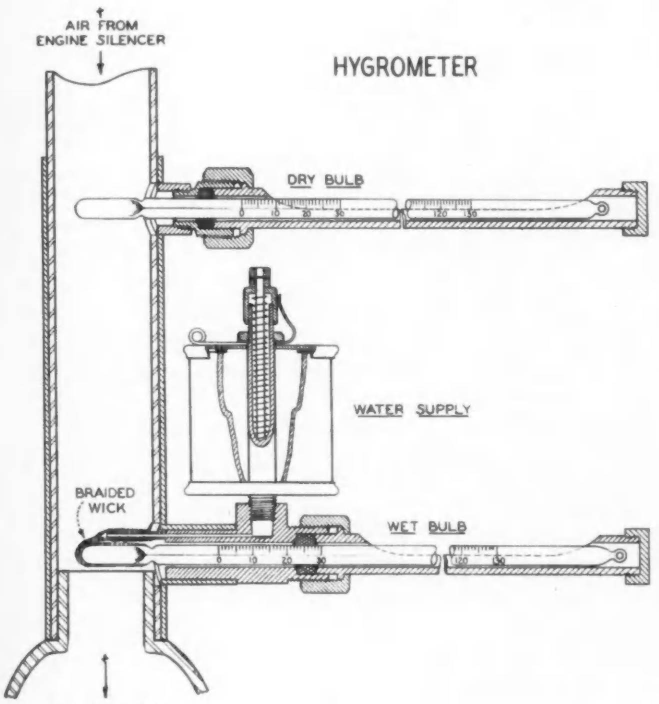


Fig. 2

the hygrometer just described, the humidity was also measured by the following analytical method:

Three calcium-chloride drying tubes were connected in series between the air supply being investigated and a calibrated wet test meter. The weight of the drying tubes before and after the measured quantity of air has been passed through provides a simple and accurate means of measuring absolute humidity. The humidity measured by the wet-bulb thermometer was found to agree closely with the values obtained by this analytical method, and no appreciable humidifying resulted from the wet-bulb thermometer wick.

Air Drier

Fig. 3 shows, in diagrammatic form, the experimental air drier used to obtain humidities lower than those prevailing in the laboratory. It consists essentially of a container for drying material, electrical heaters for dehydrating the adsorbent, and suitable control valves interposed between the unit and the engine. 5 lb. of four-mesh and 5 lb. of twenty-mesh Sila-Ka-Gel were used in two layers placed above a retaining screen. These sizes were selected as the most suitable from the small stock of the Gel on hand. In order to maintain a low humidity of 0.002 to 0.005 lb. of water per lb. of dry air, it was found necessary to dehydrate the Gel every 8 to 16 hr. of running, depending upon the atmospheric humidity. This operation was accomplished by passing compressed air from the line over the strip heaters, and thence through the drier at a temperature of 250 deg. fahr. to 300 deg. fahr. The diameter of the adsorbent container used is approximately 14 in. If it is desired to use other dimensions, a suitable low-pressure drop can be obtained by using the packed-column formula and charts given in the *Journal of Industrial and Engineering Chemistry*, Vol. 23, p. 913, 1931, or p. 744 of the first edition of the "Chemical Engineers Handbook." The ½-in. of water-pressure drop obtained in the drier described was found quite satisfactory as regards engine operation and agreed closely with the formula and charts mentioned previously.

The valves shown are for the purpose of providing control of the temperature of the air leaving the drier, and for mixing this air with atmospheric air in any desired proportions before entering the humidifier. By means of these controls the drier may be used merely as a trimmer and, therefore, will require dehydrating at less frequent intervals.

Humidifier

Fig. 4 shows the humidifier as now used on engines in the Standard Oil Co. of Calif. laboratory. It consists of a thermally insulated separating chamber provided for the purpose of accumulating and withdrawing any condensate that may occur. Near the top an indexed needle valve is provided which has both fine threads and a flat taper in order to allow easy control of the steam quantities supplied the engine air. The steam-jacketed and thermally insulated discharge tube is provided to prevent condensation of the small quantity of steam which will cause irregular operation of the humidifier.

The humidifier is made up as a unit with a section of tubing which is attached to the top of the engine silencer, and provision is made at the upper end of this tube for connection with the air drier when required. It may be noted that, in any investigation requiring the development of humidities below those normally existing in the laboratory, it has been found most convenient to establish roughly a humidity somewhat lower than desired and then trim to the desired value by means of a humidifier.

Exploratory Tests – Humidity

Unfortunately no suitable instrumentation has yet been developed for indicating, on an absolute basis, the amount of

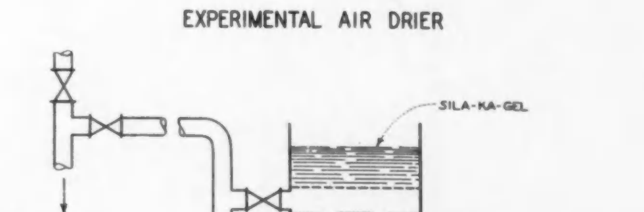


Fig. 3

knock experienced during the detonating operation of an engine. However, the conventional bouncing pin affords some measure of this variable and, in the following exploratory tests, was used as the best available means of measuring the intensity of detonation. In order to convert these knockmeter indications to equivalent octane-number changes, two reference-fuel blends differing in octane number by a known, small amount were used with each group of test fuels.

Two or more of the following fuel types were used in each of the tests: straight run, benzol-straight-run blend, cracked, leaded straight run, and octane-heptane blend.

In the first test, the compression ratio was adjusted for an approximate mid-scale position of the knockmeter when operating on a 68-octane-number benzol blend at an intermediate humidity of 0.0135 lb. of water per lb. of dry air. The compression ratio was maintained at this fixed value for all subsequent tests on this 68-octane-number group of fuels. Maximum knock mixture ratios were maintained by a slight readjustment of the carburetor as the humidity was changed due to the displacement of part of the intake air with water vapor. Knockmeter readings were then recorded over a range of humidities varying from approximately 0.003 to approximately 0.03 lb. of water per lb. of dry air.

Fig. 5 shows the results obtained in this first series of tests. It will be noted that the straight run, the cracked, and the benzol blend showed a similar response to changes in absolute humidity, whereas the heptane-octane blend and the leaded-straight-run blend were at marked variance to the other fuels.

In order to make the relations shown in Fig. 5 somewhat more significant, the changes in absolute knock, in terms of equivalent octane number, are shown for changes in humidity

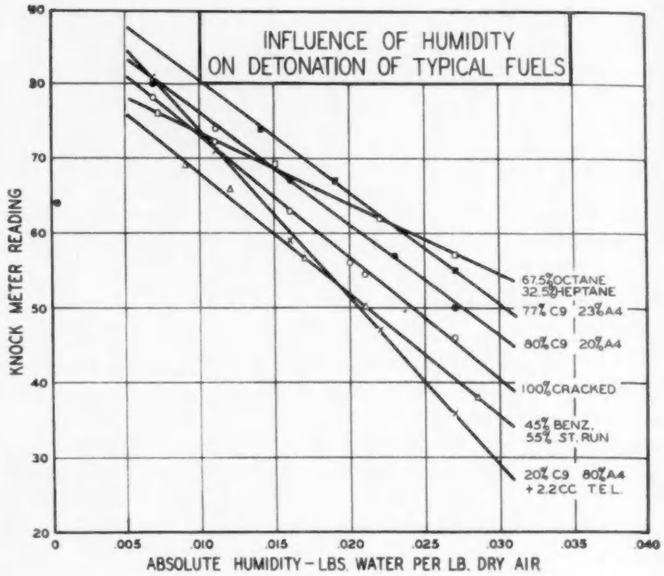


Fig. 5

from 0.002 to 0.023 lb. water per lb. dry air which is equivalent to a change in water vapor pressure of 1 in. hg.

Octane-heptane	5.1
C9 – A4	7.2
C9 – A4 plus tetraethyl lead	10.8
Cracked	7.9
Straight-run benzol	7.9

In this tabulation, the changes are expressed in equivalent octane numbers by applying the knockmeter versus octane-number correlation obtained by means of the two known reference-fuel blends, as previously described. Since the curves have slopes of the same algebraic sign, to determine the error in rating by the change in humidity indicated, it is merely necessary to subtract the value shown for the fuel in question, from the reference fuel desired. For example, the error in rating *C9 – A4* plus tetraethyl lead against unleaded *C9 – A4* would be 10.8 — 7.2, or 3.6 octane numbers.

An analysis of the data shown in Fig. 5 suggested the extension of the test to include the investigation of fuels, both higher and lower in nominal octane numbers.

Fig. 6 indicates the results obtained in a manner similar to that used in Test No. 1 but is confined to *C9 – A4* and octane-heptane blends. In this figure it will be noted that the variation in response of the two fuel types to humidity changes was negligible.

In a similar manner, Fig. 7 illustrates the relation found for similar types of fuels in the 73-octane-number range. In this case, however, the response of the octane-heptane blend was considerably less than that of the reference fuel blends.

Fig. 8 is, perhaps, of more importance than either Figs. 6 or 7, in that it indicates the source of some difficulty that has been experienced in attempting to calibrate secondary reference fuels in the higher brackets. In this range the reference fuel requires the addition of lead. In this case it will be noted that the response of the leaded reference fuels was somewhat greater than found in Test No. 1, as shown in Fig. 5, and considerably more pronounced than for the unleaded reference fuels used in the tests for which results are shown in Figs. 6 and 7.

A critical review of the data contained in Figs. 5 to 8, inclusive, will indicate that the differences in each case are not in exact agreement. In this connection attention is called to the fact that, in converting these results to an equivalent octane-number basis, an error of 10 to 20 per cent is possible, since the equivalent octane-number change was derived from

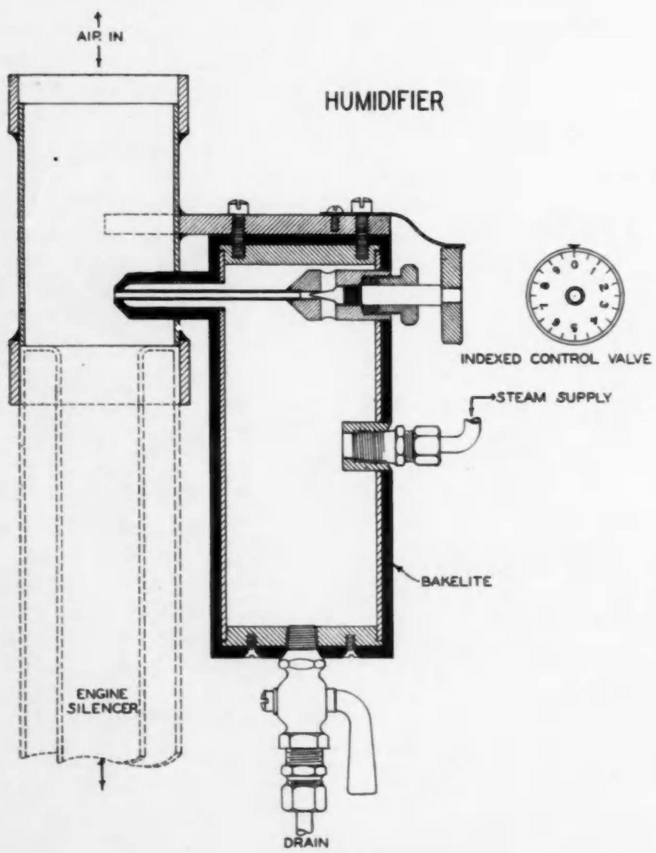


Fig. 4

the knockmeter sensitivity as indicated by the two reference-fuel blends, differing by a "known" amount of 1 or 2 octane numbers. If a 0.2-octane-number error had been made due to an error in mixing these reference fuel blends, an error of 20 per cent in all other equivalent octane-number values on the same chart compared to the values on another chart would result in the case of the 1-octane-number bracket, and a 10 per cent error in the case of the 2-octane-number bracket.

Due to the apparent sensitivity of certain types of fuels to knock intensity and, therefore, perhaps also to bouncing-pin settings, the relative values shown in the various figures have been found in subsequent tests to vary somewhat in direction as well as magnitude. As an illustration, tests of a similar nature to those for which the values have been shown have indicated slightly different relations, particularly as regards the relative response of octane-heptane and straight-run fuels. In the latter tests, somewhat different conditions may have arisen due to a slight difference in the method of establishing standard knock and also to the bouncing-pin settings occasioned, in part at least, by the change in knock intensity.

Exploratory Tests – Dry-Air Pressure

It has been assumed that the influence of humidity on detonation is merely the result of changes in dry-air pressure caused by the change in water-vapor pressure. If this assumption were true, detonation should be influenced to the same extent by a given change in dry-air pressure whether that change is caused by a variation in either barometric or water-vapor pressure. The following tests, covering a range of barometric pressure, were made to compare the influence of the two methods of changing dry-air pressure.

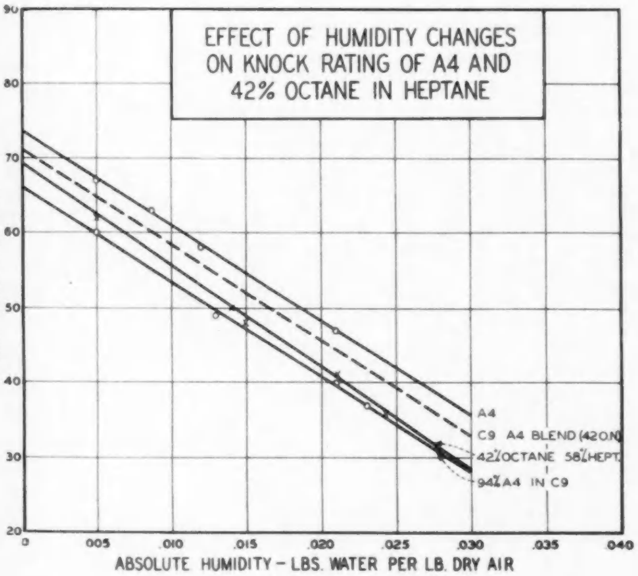


Fig. 6

The laboratory engine was equipped with surge tank and means for controlling the intake-air pressure to any predetermined value below the prevailing atmospheric pressure. In view of the relatively small influence of normal changes in water-vapor pressure on the dry-air pressure, the changes in barometric pressure required for an equivalent effect by the throttling of the intake air were limited to the order of 2 in. hg. depression.

With the throttling set-up and with the humidity maintained constant, tests were made using the same types of fuels as employed in the humidity tests. As in the case previously reported where the humidity was changed, variations in knock intensity were determined by means of the knockmeter. The testing of two normal reference-fuel blends again provided a means of establishing the sensitivity of the knockmeter to permit the evaluation of the results in terms of apparent octane-number changes brought about by differences in intake-air pressure.

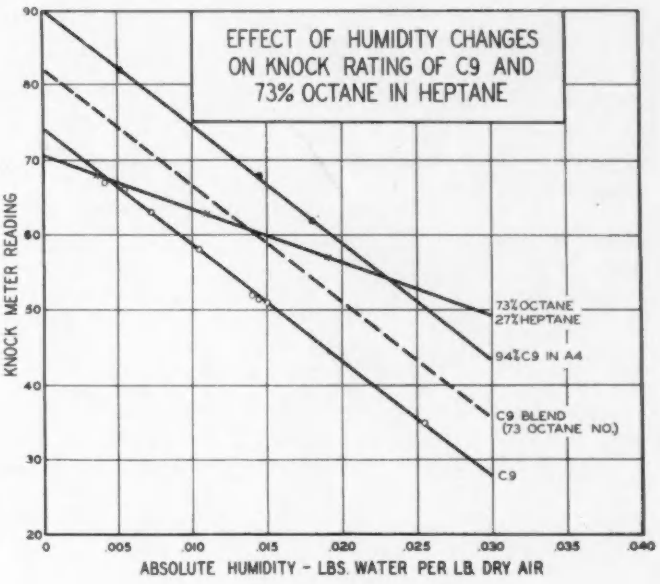


Fig. 7

Fig. 9 represents the data obtained in these tests, plotted as changes in intake-air pressure and corresponding changes in apparent octane number. The curves shown in Fig. 9 yield some rather interesting relations. It will be noted that the leaded reference fuel was least affected, the benzol blend moderately affected, whereas the cracked, straight-run, and octane-heptane blends responded most to changes in intake air pressure. In order to make the relations shown in Fig. 9 more comparable with the data from Fig. 5 previously tabulated, the changes in apparent octane number for similar changes in dry-air pressure of 1 in. hg. are shown below:

Octane-heptane blend	3.4
*C*9 – *A*4	3.4
*C*9 – *A*4 plus tetraethyl lead	2.5
Cracked	3.4
Straight run-benzol blend	2.8

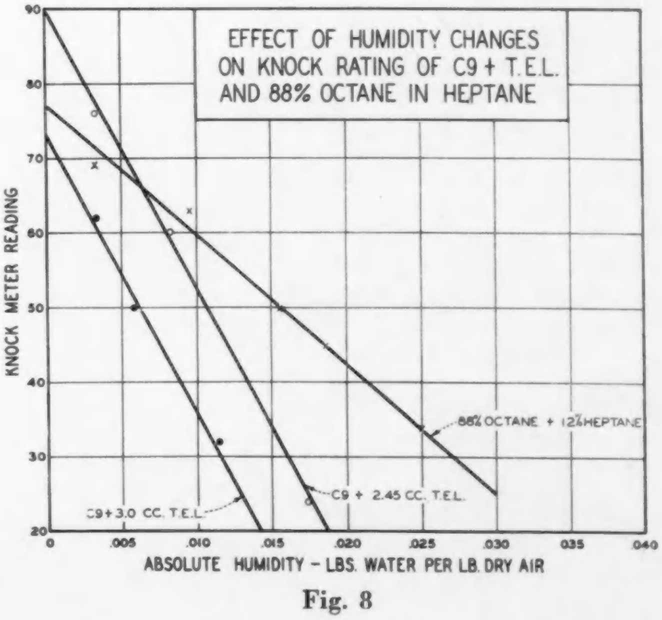


Fig. 8

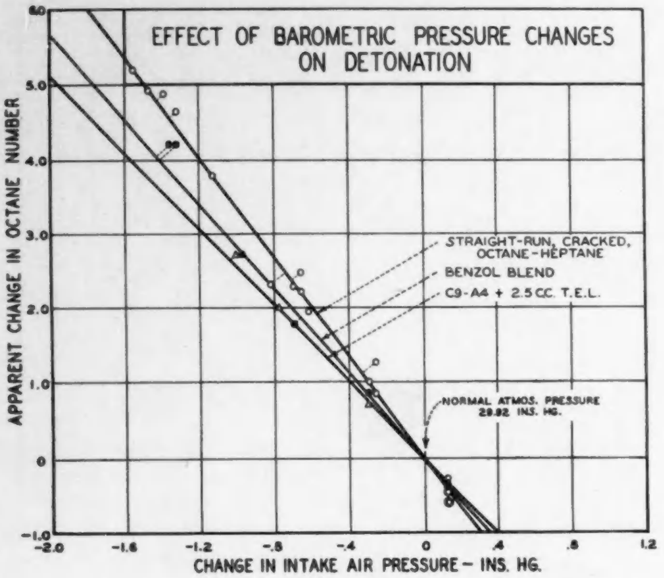


Fig. 9

A comparison of the tabulated data taken from Figs. 5 and 9 illustrates very definitely that dry-air-pressure change alone does not explain the effect of humidity on detonation. For instance, attention is called to the relation of the effect on the leaded reference fuel to the effect on other fuels tested, as shown in Fig. 9. Had humidity been effective merely from the standpoint of the replacement of air by water vapor with a resultant decrease in dry-air pressure, the relation shown on Fig. 5 for this type fuel would have been reversed. Therefore, the net effect of humidity changes on leaded-reference-fuel blends is apparently the difference between the effect of water vapor on combustion and the effect of diminished air pressure. It may be concluded, therefore, that correcting for humidity on the basis of dry-air pressure alone will not eliminate the necessity for humidity control.

In normal knock-testing work, the largest error is likely to occur when calibrating straight-run reference fuels plus tetraethyl lead against octane-heptane, or when bracketing leaded fuels with unleaded straight-run reference fuels. If the humidity were abnormally high or low at the time the reference fuel was calibrated against octane-heptane, all subsequent tests using this calibration would be in error, even though average humidity prevailed and the test fuel was similar in type to the reference fuels.

Tests by Direct Matching

As previously mentioned some uncertainty exists as to the accuracy of the results obtained when using the bouncing pin as an absolute knock indicator. Tests were made, therefore, using the fuels displaying the greatest relative effect, to determine by the conventional matching method the relative importance of dry-air pressure and humidity changes.

Using a blend of reference fuel *C9* and *A4* of such proportions that, when 2.5 cc. of tetraethyl lead per gal. were added the nominal octane number was 70, two sets of tests were carried out. In each case the leaded fuel was matched directly by means of unleaded reference fuels *C9* and *A4*. With a constant humidity of 0.008 lb. of water per lb. of dry air the matching octane number assigned from the published calibration of these two reference fuels was 70.3 at a barometric pressure of 30.22 in. of mercury. With this pressure but with the humidity raised to 0.029 lb. of water per lb. of dry air, equivalent to a reduction in dry-air pressure of 1 in. hg., the matching octane number was 73.3 or a difference of 3 octane numbers. This difference compares favorably with the difference of 3.6 obtained from Fig. 5 for similar fuels.

In the second test the dry-air pressure again was reduced by 1 in. hg., but in this case by throttling the intake air rather than by raising the humidity. The octane number obtained by matching was 70.8, a difference of 0.5 octane number. This value compares with the difference of 0.9 octane number obtained from Fig. 9.

A third test consisted of calibrations of a leaded straight-run reference fuel, made in the normal manner except that controlled humidities of 0.004, 0.019 and 0.036 lb. of water per lb. of dry air were used. The results of these calibrations are shown in Fig. 10.

Apparently in place of a fixed calibration curve, a family of curves can be drawn, one for each humidity existing at the time of calibration. Definite changes in our calibration curves, until now unexplained, have occurred from time to time when checking the calibration of our secondary reference fuel against octane-heptane. Fig. 10 explains, to a large measure, the cause of the discrepancies previously noted.

Attention is called to these tests particularly since they are indicative of normal changes in atmospheric conditions and were made with normal fuels. Humidity, therefore, is a variable of sufficient importance to warrant adequate control in the laboratories engaged in the evaluation of knock rating of fuels.

Humidity Control in Routine Testing

In view of the need for humidity control even in routine testing of conventional fuels, the question arises as to whether or not the equipment described is so complex as to demand the attention of one especially skilled in the manipulation of such control equipment. It was considered advisable, therefore, to determine whether the average engine operator, such as used in the majority of refinery control laboratories, would have any difficulty using the equipment described.

For this test, one of the C.F.R. operators from our control laboratory who had had no previous experience with special equipment was used. No instructions were furnished other than a brief written outline of the method of control and a list of wet-bulb temperatures required with various dry-bulb temperatures between 70 and 90 deg. fahr. to maintain a nominal humidity of 0.0135 lb. of water per lb. of dry air. With the humidity thus controlled by the operator, each of the following four fuels were rated twice: cracked, leaded straight-run, benzol blend, and commercial-iso-octane blend. The maximum difference between ratings of the same fuel was 0.4 octane number. The humidity was then controlled by a person most

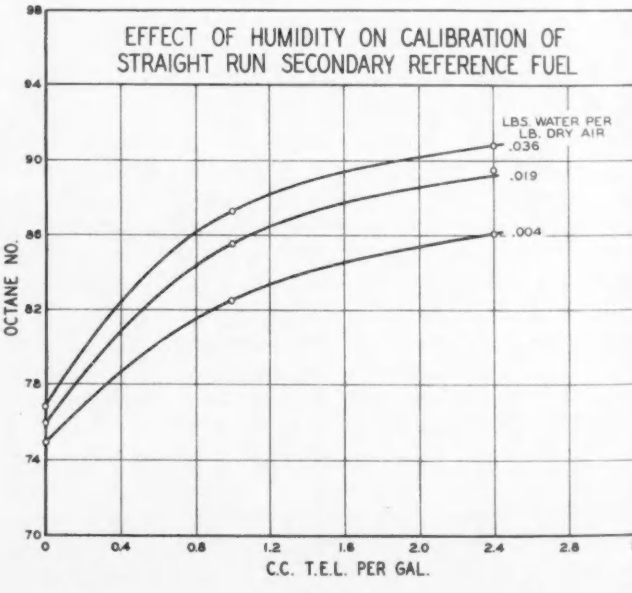


Fig. 10

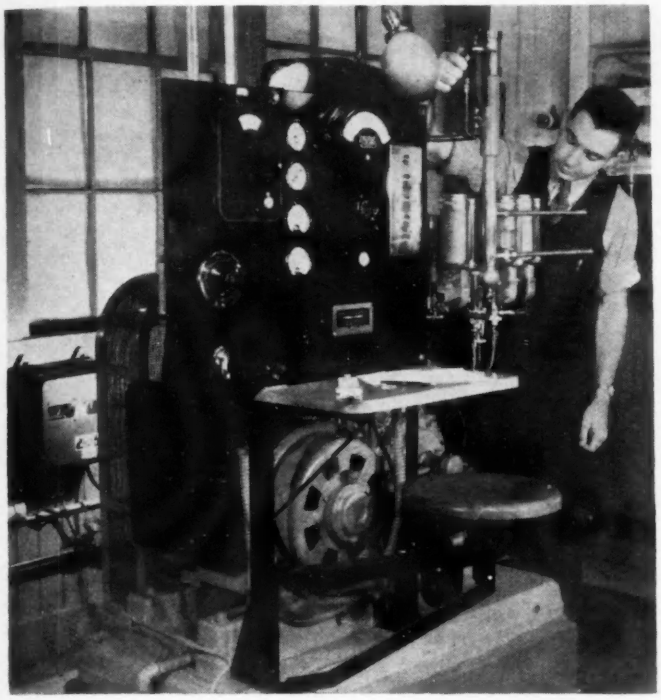


Fig. 11 – Standard Oil of Calif. Humidity-Control Equipment

familiar with the operation of the drier and humidifier, first at 0.003 and then at 0.025 lb. of water per lb. of dry air while the operator re-rated the four fuels at the two humidities in the normal manner. In this case, however, the values at the two humidities differed by as much as 3.9 octane numbers.

The improvement in accuracy obtained and the lack of operating difficulties demonstrated that the equipment used is satisfactory for this type of work.

Various methods of controlling humidity are available, each of which has its own advantages. The method described has proved very satisfactory and should be found so in other laboratories in which the normal range of humidity does not exceed a value of more than about 0.015 lb. of water per lb. of dry air.

Fig. 11 shows the unit with its appurtenances attached to one of the engines in the laboratory of the Standard Oil Co. of Calif. It will be noted that it is compact in arrangement and is controlled conveniently. Attention is called to the Micromax indicator mounted on the wall behind the engine. This device controls the mixture temperature by means of a thermocouple placed in the intake manifold of the engine in close proximity to the standard laboratory thermometer. This automatic control has been found very satisfactory since it is able to react to smaller changes in mixture temperature than can be observed by the operator. This advantage is of particular importance when changing from one fuel to another if there is a difference in their latent heats. The time saved by this equipment is very large in laboratories where a wide variety of fuels are tested in a normal day's work.

Standard Humidity

An analysis of Fig. 1 indicates that, at the time of year in which the C.F.R. Motor Method was established originally as a means of rating fuels in the laboratory in accordance with their road performance, the humidity was probably in the range of 0.013 to 0.014 lb. of water per lb. of dry air. Specific data on this question are not available, but the average humidity prevailing at the same time of year in the various laboratories participating in the cooperative calibration of reference fuel $C9$, was 0.0135 lb. of water per lb. of dry air. This value was, therefore, established as the approximate mean around which humidities were varied in the exploratory tests herein discussed and still remains as the standard humidity under which all knock ratings in this laboratory are made.

Summary

In summary, it may be stated that for refinery control it is possible that for some normal gasolines bracketed with straight-run reference fuels, providing the same are calibrated under standard humidity conditions, no great error will result with the humidity varying over the normal range. However, with the increase in use of special-type fuels for reference-fuel calibration and especially for fuels containing tetraethyl lead, some form of humidity control is required. The exact humidity to be used as standard has not yet been determined but satisfactory reasons appear for choosing 0.0135 lb. of water per lb. of dry air, and its adoption as standard in all knock-rating tests is hereby suggested.

Discussion

Author's Findings Confirmed with Different Apparatus

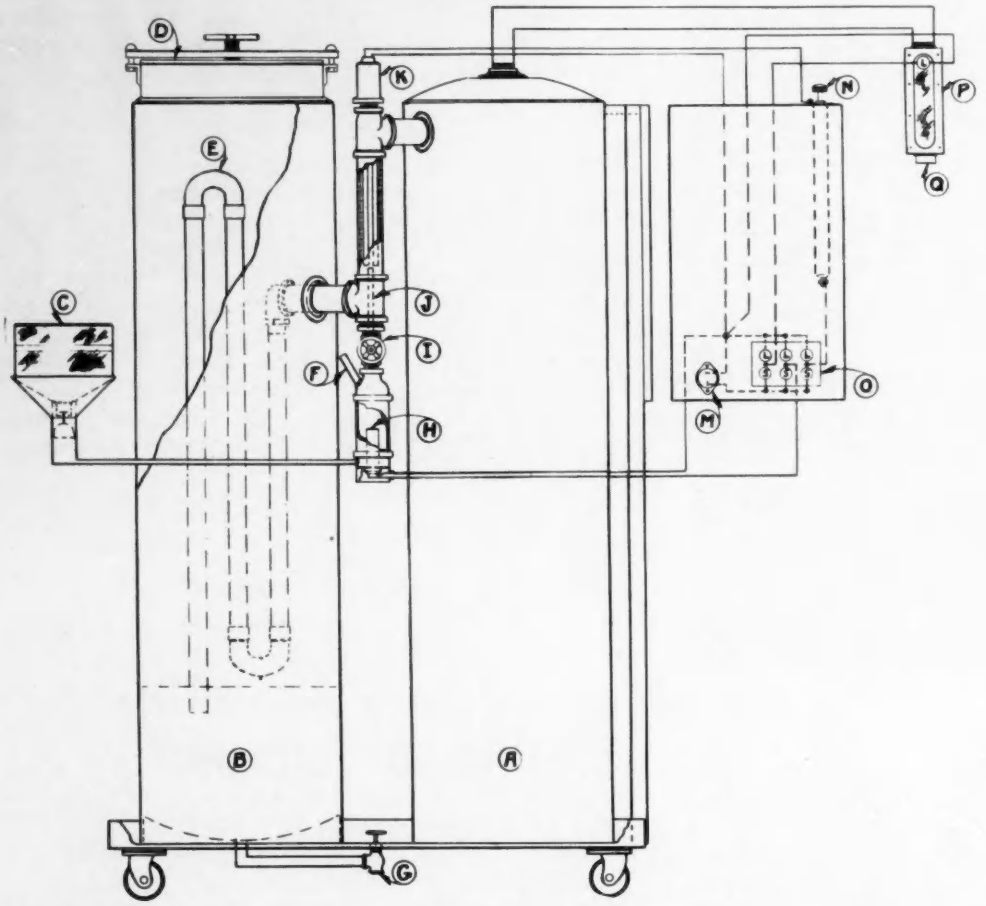
– Earl Bartholomew
Ethyl Gasoline Corp.

MR. MACGREGOR has called to our attention in a most satisfactory manner the importance of a variable in knock testing which heretofore has not received a great amount of attention notwithstanding our desire to improve the reproducibility of knock ratings in the same and in different laboratories. In our own research laboratories we have observed over a period of several years that the effectiveness of lead with respect to the commonly used reference fuels increases with an increase in the moisture content of the air used by the knock-testing engine. Fuel-testing laboratories of the Ethyl Gasoline Corp. are located at Yonkers, New York, Detroit, Kansas City, Tulsa, and Baton Rouge. In the rating of check samples sent periodically to all laboratories there has been a consistent tendency for the Baton Rouge laboratory to find less lead required to produce a given octane number, measured in terms of the secondary reference fuels. Higher absolute humidity at Baton Rouge is the most obvious difference in the testing conditions at that point hence there is reason to believe that the experience of the testing laboratories confirms the observations made in the research laboratories.

It could be argued perhaps that seasonal variations in humidity are self-correcting with respect to car performance since any difference in the rating on the laboratory engine also would be reflected on the road. To some extent this reasoning is true but, during the summer months and to a lesser degree at other seasons, the absolute humidity is subject to violent fluctuations from day to day; furthermore, gasoline rated at a given refinery may be consumed in another part of the country at a considerably later time. The only satisfactory solution is the standardization of humidity conditions as Mr. MacGregor has suggested.

Our studies of the variation of absolute humidity in different parts of the country during the year have led us to the conclusion that Mr. MacGregor's suggestion of standardization at 0.0135 lb. of vapor per lb. of dry air is approximately the correct average figure. It has been observed in our laboratory that, as the atmospheric humidity is varied over a wide range, there may be a considerable change in the degree of correlation of ratings determined by the bouncing pin and by the ear. This variation further complicates the problem and suggests the desirability of an extensive study of all phases of the humidity problem with different types of instrumentation in as many laboratories as possible.

Variations in humidity in our original work were obtained by the addition of moisture to the air in the knock-testing engine room on days of extremely low outside temperature, but the limited flexibility of this procedure soon gave way to apparatus which used ice for cooling and partial dehumidification, activated alumina for additional dehumidification, steam for humidification, and electric heaters for reheating. This apparatus made possible a reduction of humidity to almost the zero point but had the disadvantage of large size and the necessity for regeneration of the activated alumina. It was considered that for ordinary purposes a minimum humidity of that corresponding to a dew point of 32 deg. fahr. at atmospheric pressure would be entirely satisfactory, hence the apparatus



A—Surge chamber.
B—Ice chamber.
C—Water bottle.
D—Air orifice (inlet)
E—Air-cooling coil.
F—Steam generator vent.
G—Water drain valve.
H—Heater for steam generator (110 volts, 250 watts).
I—Steam metering valve.
J—Steam discharge pipe.
K—Air heater (110 volts, 2000 watts).
L—Lights.
M—Outlet for 110-volt current.
N—Rheostat for air heater.
O—Electric switches and pilot lights.
P—Hygrometer housing.
Q—Air outlet to engine.
S—Electric switches.

Fig. *A* – (Bartholomew Discussion) Diagrammatic View of Air Conditioner

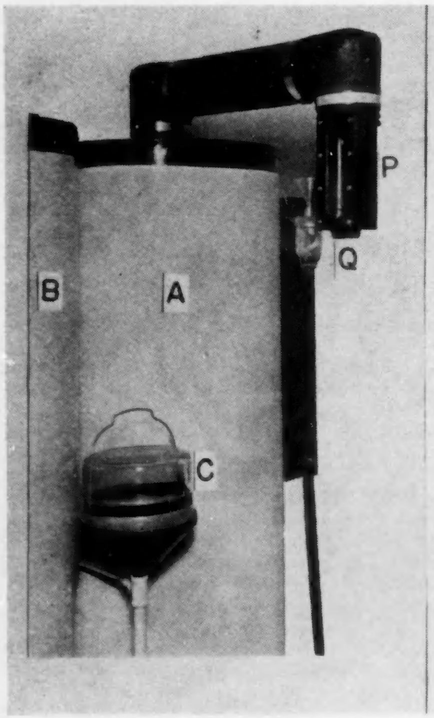


Fig. *B* – Thermometer Chamber, Water Reservoir for Wet-Bulb Thermometer, and Water Container for Steam Generator

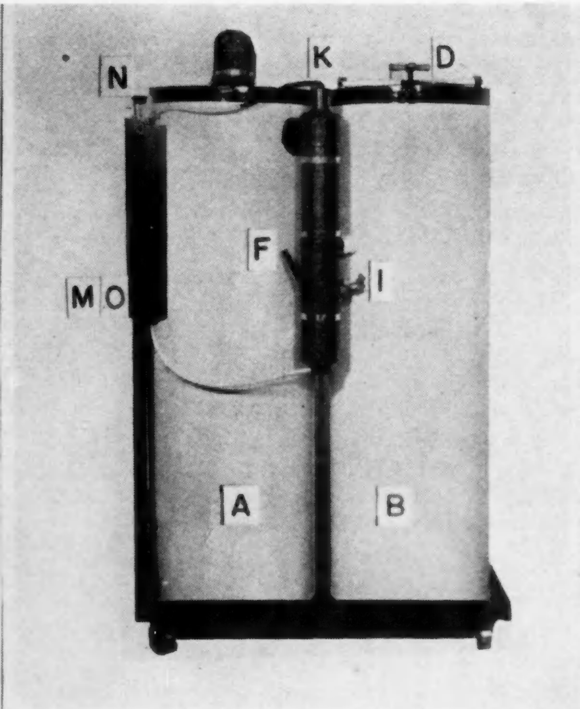


Fig. *C* – Side View of Conditioner Showing Steam Generator and Heater Chamber

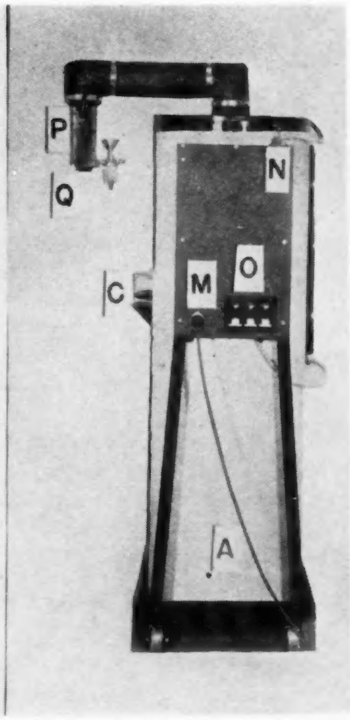


Fig. *D* – End View of Conditioner Showing Electric Control Panel Which Supports Inclined-Tube Manometer When Used

Figs. *B*, *C* and *D* (Bartholomew Discussion) – Letters Refer to Legend in Fig. *A*

shown diagrammatically in Fig. *A* was designed. In the figure, *A* and *B* are 52-gal. cylindrical tanks which are enclosed in larger concentric casings, the space between being filled with expanded mica insulation. Cylinder *B* is normally filled with cracked ice and cylinder *A* serves as a surge tank, it having been found by experience that surge chambers having too small a volume may reduce considerably the volumetric efficiency of single-cylinder engines and introduce carburetion problems.

Air enters the ice chamber through a removable round-edge orifice *D*, passes downward through the ice, flows through the copper-tube assembly *E* and enters chamber *J* where its humidity and temperature are brought to the desired values before passing into surge chamber *A* and out to the engine through opening *Q*. Steam in excess of the quantity ever required for humidification is generated at atmospheric pressure in chamber *H* which is vented to the atmosphere through tube *F*. The quantity of steam added to the air is controlled by gate valve *I* whose adjustment is not critical since the specific volume of steam at atmospheric pressure is high and the pressure differential across the valve is only the 1 to 2 in. of water induced by the round-edge orifice *D*. Steam is discharged into the engine air between the blades of heater *K* which is similar to that used for mixture-heating on the C.F.R. engine. A carbon pile rheostat *N*, similar to the one used on the C.F.R. engine and mounted on the rear of the control panel, is used to regulate the heating of the air.

Before entering the engine, the air passes through chamber *P* which is equipped with a glass window and houses the wet- and dry-bulb thermometers. The means for supplying water to the wet-bulb thermometer is somewhat similar to that employed by Mr. MacGregor. In this case the water reservoir is provided with a conventional gasoline filter having a removable glass bowl which can be seen in Fig. *B*.

Thermometer chamber *P* is provided with a small light. This light, as well as the steam generator and air heater, operate on 110-volt current provided by a multiple connection at the mixture-heater plug on the rear of the C.F.R. engine control panel. This arrangement prevents the steam generator and air heater from being left in operation after the engine has been stopped.

Water is maintained at a constant level in the steam generator by the chicken-feed action of the inverted glass water tank *C* which is equipped with a spring-loaded valve on the cap similar to that employed on fuel containers for kerosene stoves. The steam generator consumes 250 watts at all times. The water tank has a capacity for approximately 20 hr. of operation.

The ice chamber is fitted with an air-tight removable cover hence, by the addition of a suitable manometer, the round-edge orifice may be used for the determination of air consumption if the vent of the steam generator is trapped properly.

The temperature of the air leaving the ice chamber is between 33 and 34 deg. fahr. at the point of measurement. Although its temperature is increased somewhat by the time it reaches the dry-bulb thermometer, the humidity determinations made by means of the wet- and dry-bulb thermometers indicate that the air has a humidity corresponding to saturation at 32 deg. fahr.

Fig. *C* shows a side view of the conditioner with steam generator and heating chamber. Fig. *D* shows an end view of the conditioner with electric control panel which also supports an inclined-tube manometer when used.

There are certain definite advantages in dehumidification by means of ice. Regardless of the conditions under which the knock-testing engine is operated or of variations in the temperature and humidity of the air in the knock-testing engine room, the air leaving the ice chamber always is saturated at 32 deg. fahr. This condition insures a minimum amount of adjusting of the steam-control valve and heater rheostat during the day. The absence of the need for regeneration perhaps is balanced by the necessity for charging the ice chamber. One filling lasts for approximately two days, the cost being approximately 30 cents per day.

Our own experience confirms Mr. MacGregor's in that an operator without previous experience with the conditioner is able to make ratings with controlled air temperature and humidity without any assistance. It appears that large variations in the range of wet- and dry-bulb temperatures met in different parts of the country will make desirable different designs of apparatus to meet particular local conditions.

Discussion of "Cetane Numbers - Life Size"

(*Continued from page 242*)

The lag of as much as 0.013 in. required by the injector to make contact is undesirable. The author claims that the error caused by this delay and that caused by gas passage connecting the combustion-chamber to the pickup diaphragm are opposite and about equal, so they cancel out. The writer's estimate is that the time taken by the needle lift is much longer and will vary with the viscosity of the fuel. We have found that much thicker diaphragms are able to actuate a pickup (in fact our pickup responds perfectly by simply touching the cylinder-head with it), which would make the water-cooling of the diaphragm superfluous.

The author no doubt also has made the observation that high-viscosity fuels like F_4, F_5 and the *B* blends gave much higher ratings (frequently by more than 10 cetane number) with the submarine engine than with the C.F.R. engine, using the delay method in both. Can he advance some explanation for this fact?

Differs with Author's Conception of Ignition Quality

—C. C. Moore, Jr.
Union Oil Co. of Calif.

In agreement with the author, we feel that ignition quality may be defined advantageously as a specific property of Diesel fuel very largely dependent on the chemical composition and but slightly modified by the physical properties except in so far as these are influenced by the chemical composition. Lt.-Commander Good's procedure in using a two-cycle engine for testing fuels intended for use in a two-cycle engine is excellent, particularly if relative ratings cannot be established in a standard test engine.

The oscillographic apparatus described is extremely interesting and, in our opinion, is a major contribution to the tools which we have to study the ignition and combustion characteristics of Diesel-engine fuel oils. The ability to obtain a photographic record of the rate-of-pressure-change curve with this apparatus is of special interest. It would seem off-hand, that better indication of the beginning of ignition could be had from this curve than from the cylinder-pressure curve as used by Lt.-Commander Good. In the case of the high-cetane-number blends especially, it appears somewhat difficult to locate a break in the pressure curve corresponding to the start of ignition. Comparison of the rate-of-pressure-change curves when fuel is injected and when it is not should enable this point to be determined with even greater accuracy.

Lt.-Commander Good states in brief that ignition quality is measurable only through its effects and should, therefore, be measured in full-scale engines in field service. We also judge that he believes computed combustion knock, as determined in a full-scale engine, is a preferred method of expressing the somewhat elusive property of how a Diesel-engine fuel oil burns. Although we recognize fully the value of Lt.-Commander Good's work, we cannot subscribe wholeheartedly to his major theses.

Ignition quality in a Diesel-engine fuel oil is a direct function of the chemical composition and an indirect function of the physical properties of the fuel, the design of the engine, the physical condition of the engine, the operating conditions, the atmospheric condition and, in some measure, the technique of the operator.

Our experience with the commonly available cylinder-pressure indicators leads us to question the degree of accuracy with which the explosion-pressure rise can be determined. In this connection it is not clear to us why the explosion-pressure rise in Fig. 6 of Lt.-Commander Good's paper was taken at the indicated point and might not just as accurately be taken as some 50 per cent greater than the indicated value. The formula for the calculation of computed combustion knock can be written:

$$\text{Computed Combustion Knock} = \frac{(\text{Explosion-Pressure Rise})^2}{\text{Degrees Crank Angle} \times 10{,}000}$$

Any error in the explosion-pressure rise is magnified by being squared and we question if the degrees crank angle through which the engine travels during the time the explosion-pressure rise occurs can be measured any more accurately than can the ignition-delay period. The degrees crank-angle travel during the explosion-pressure rise is ordinarily of the same or smaller magnitude than the ignition delay in degrees crank angle. This measurement alone, therefore, introduces as much or more error than is inherent in the ignition-delay method.

In writing this discussion we have been somewhat handicapped by a lack of exact knowledge of how the value for explosion-pressure rise was determined. It is felt that the value of the paper would be enhanced if a detailed explanation of this phase were given.

We are somewhat surprised that better differentiation of the high-cetane fuels is given by the large, relatively slow engine used than by the relatively fast, small laboratory engine. This result is not in accord with our findings or those reported by other investigators.

From the oil refiner's viewpoint, cetane numbers must be reduced from "life size" to laboratory size. Lt.-Commander Good's paper shows the need for correlation between the life-size test and the laboratory test and, in reducing from life size to laboratory size, every precaution should be taken to insure that an adequate correlation has been established.

We are fully cognizant that in the case of "doped" fuels, the physico-chemical indexes are ordinarily valueless. In general, however, the sale and use of "doped" fuels have not yet become extensive, and as a ready means of differentiating between good and bad fuels we feel that physico-chemical indexes have a distinct place. Future investigations undoubtedly will result in improved correlation between physico-chemical indexes and service performance.

Aircraft-Engine Installation Vibration Problems

By John M. Tyler

Hamilton Standard Propellers, Division of United Aircraft Corp.

THE reduction of vibration in aircraft structures is becoming a major problem in aircraft design. The increasing emphasis placed on vibration studies is the result of three important influences: (1) the increased loading of the structures, (2) the broadening of the operating-speed range obtained with controllable propellers, and (3) an increase in demand for more passenger comfort.

In the past structural vibrations have been given attention mainly because of the danger of failure. At present to obtain the required comfort, the vibration studies must be carried out to a much greater degree of refinement.

The vibrations of the aircraft engine as a whole may set up vibrations of the whole airplane, but the vibrations of parts of the engine with respect to each other are not noticeable in the cabin; in fact, they are difficult to detect except by the use of special instruments.

A study of the vibration characteristics of an engine-propeller installation involves (1) a determination of the vibration spectrum of the installation, that is, a determination of the frequencies of all of the modes of vibration, (2) a study of the exciting forces set up within this combination, and (3) the operating conditions, that is, the operating speeds and the power required at these various operating speeds. From these data the engineer can predict the vibration characteristics of the installation.

THE status of the vibration work in the aircraft industry is a good index of the status of the industry as a whole. A few years ago vibration was looked upon as a very dangerous phenomenon which might, at any moment, cause a failure of some important structural part of the airplane, engine, or propeller. Today most of the work being done on vibration is aimed to increase the comfort of airplanes. We have now reduced the vibrations in standard equipment to a point where they are no longer dangerous, and we now have time to attack the problem of the transmission of vibrations from their source to the passenger seat.

For various reasons, there have been more danger and discomfort in aircraft than in other vehicles. The necessity for designing to save weight has cut down the margin of safety of the aircraft parts. Thus a vibration which doubled or tripled the design stress in an airplane part probably would cause failure, whereas the corresponding part in an automobile would have a sufficient margin of safety to stand the additional stress.

Until recently discomfort in airplanes has been attributed largely to propeller disturbances. The opinion has been that the noises and vibrations not due to the propeller are a negligible portion of the whole. But now, with the appearance of some comparatively comfortable airplanes, many of the old illusions are being dispelled, and the airplane and engine manufacturers are tackling the problem of engine roughness all by itself.

The problem of passenger comfort in airplanes is allied closely with the problem of human reaction to various vibration sensations. The scope of this paper does not permit more than a very brief sketch of the work done in this field.

Several experimenters have studied the reaction of human beings to special sections of the vibration spectrum. Professor LaMair, at the University of Lyons, introduced the vibrating chair and established that the edge of comfort was at $0.4 \times 10^{-6} AN^3$ where A and N are amplitude and frequency. Prof. H. M. Jacklin[1,2] at Purdue has carried this work further and has established curves which locate the edge of comfort for various types of motions. These studies show that the exponent in Professor LaMair's AN^3 factor should not always be 3, but is different for different conditions depending on direction of motion, frequency, and other factors.

It also has been pointed out that the structure of the viscera has been subjected to one band of vibrations for ages – the frequency of the leg motion in a normal walk and trot. The elastic suspension of the internal organs from the walls of the viscera has, by the processes of evolution, been so adjusted that resonant vibrations do not occur in the range of walk and trot frequencies. Just below and just above this range there are frequency bands that set the internal organs into motion, causing them to dump excessive amounts of their secretions into the intestines, and the familiar nausea of seasickness or air sickness results. Nausea is connected more definitely with frequencies below the walk and head-

[This paper was presented at the National Aircraft Production Meeting of the Society, Los Angeles, Calif., Oct. 16, 1936.]

[1] See Research Bulletin No. 44, Engineering Experiment Station, Purdue University, May, 1933; "Riding Comfort Analysis," by H. M. Jacklin and G. J. Liddell.

[2] See S.A.E. TRANSACTIONS, October, 1936, pp. 401-407; "Human Reactions to Vibration," by H. M. Jacklin.

aches and excessive fatigue with frequencies above the trot. A fast-trotting horse will give an inexperienced rider plenty of vibration just above the human-trot frequency.

Another phase of the effects of vibrations on human beings is that of steady accelerations or the low-frequency vibrations below the walk frequency, such as elevator motions, streetcar swaying, boat rolling, and airplane motions in bumpy weather. These motions put a steady strain on the supports of the organs in the viscera and produce much the same effect as the motions that set the internal organs into resonant vibrations.

Still another very important factor is the psychological reactions of the person being subjected to the vibration. It is well known that mental reactions are closely related to the reactions of the organs in the viscera and vice versa. For example, the physical effect of such emotional reactions as fear, anger, hate, love, and so on, are the result of glandular reactions in the viscera. Thus the fear of seasickness and constant dwelling on the subject can produce the same effect on a human being as the motion of a boat on the water. Good examples of this effect are the people who actually become seasick on a boat before it leaves the dock. The same reactions are noticed in air travel, but the airplane does not take as long to leave its dock.

It is well known that passenger sickness in rough air sometimes can be avoided by reducing airplane cruising speeds. This reduction increases the angle of attack and reduces the violence of low-frequency pitching by moving onto the flat portion of the center-of-pressure curve.

It is obvious from the foregoing discussion that no single formula can be used to describe the effects of vibrations on human beings. The fact is that there is a wide variation in the reactions of individuals to vibrations. Open-sea sailors often cannot take the seas in the English Channel and vice versa. It is possible, however, to establish relationships that hold for specific frequency bands. The characteristics of the viscera, the balance reactions, and nervous strain determine the human response to low-frequency motions; the elasticity and insulation at the joints seem to be important in the next band; and the circulation system seems to respond to high frequencies in some relationship similar to AN^3.

Now, to return to the mechanical phenomenon of vibrations, we find that the great danger from vibration in aircraft structures has been due mainly to the fact that they were such an unknown quantity. Designers did not have enough information available to calculate the speed at which they would occur or the stresses that would be produced. Then, after the experimental engineers had the physical objects to work with, the only way they could tell whether a vibration period would be dangerous was to try it out and see if anything broke. If failures occurred, it was definitely dangerous.

After a few rather serious accidents due to vibration a few years ago, an impression became prevalent that all resonant vibrations were dangerous. In specifications the phrase, "no resonant vibration frequencies shall exist within the operating-speed range," became familiar to all aircraft manufacturers. When vibration analysis became accurate enough to show the location of all of the resonant frequencies, it became apparent that there were so many resonant frequencies within the operating range that if resonance were to be avoided the range must be cut to only a few r.p.m. This range would give practically a constant-speed installation. With controllable propellers coming in at that time, it became apparent that the operating-speed ranges would be much wider instead of narrower. More accurate analysis later showed that many of the resonance conditions were not dangerous. This phrase now is interpreted generally to mean that there shall be no dangerous resonance conditions in the operating-speed range.

Excitation

There are three main sources of exciting forces that set parts of the airplane structure into vibration: the engine, the propeller, and the aerodynamic disturbances set up in the air flowing past the airplane. The engine and propeller disturbances have received a great deal of careful study and are fairly well understood, but the aerodynamic disturbances are understood in a qualitative sense only. Their physical dimensions are at present almost entirely unknown. The reason for this lack of knowledge is that direct observation or measurement is possible only in a full-scale wind tunnel.

Engine Excitation

There are two sets of exciting forces in the engine: the power impulses and the unbalanced mass or inertia forces. The power impulses are something that we cannot very well do without in an internal-combustion engine. When the power impulses or torque curves for all of the cylinders of an engine are added together, the resultant torque curve is a wavy line with a peak at the point where each cylinder peaks. This wavy line shows the torque variation of the engine. If we had enough cylinders or if we would round off the peaks of the torque curves, we could reduce materially the amplitude of torque variation, but it would be practically impossible to eliminate the torque variation entirely.

This torque variation or exciting torque is applied to the engine about the crankshaft axis. The torsional vibration of the engine in response to this excitation will be dependent on two other factors: the distribution of the mass of the engine and the flexural characteristics of the mounting. These factors will be discussed later in the paper.

Control of the torque variation or exciting torque from a given engine is practically impossible. There are wide differences between the torque variation of a turbine and of an Otto-cycle engine, and there is an appreciable difference between an Otto-cycle engine and a compression-ignition engine but, for a given Otto-cycle engine the torque variation is a function of the number of cylinders and the power output only. Of course, it is possible to retard the spark so far or to slow down the flame travel to such an extent that the peak of the torque curve is rounded off and the torque variation correspondingly reduced, but this method would mean throwing away an appreciable percentage of the engine power and I am assuming that engines always will be designed for maximum efficiency.

The torque variation for a given engine is directly proportional to engine torque. This statement can be expanded to include all engines with the same number of cylinders: The torque variation for all N-cylinder engines is proportional to the engine torque. An abundance of experimental and analytical data is available for use in calculating the torque variations for both Otto- and Diesel-cycle engines.[3,4,5,6]

Control of the mass balance of the engine is a different matter. Engine balance is decidedly a controllable factor. The inherent balance of an engine is determined by the kinematic relationships of the engine parts, that is, the ar-

[3] See "A Treatise on Engine Balance Using Exponentials," by P. Cormac, Chapman and Hall, Ltd., London, 1923.

[4] See *Luftfahrtforschung*, July 24, 1929; "Torsional Vibration in Vertical Engines," by Albert Stieglitz.

[5] See *Journal of the Aeronautical Sciences*, February, 1936; "Harmonic Analysis of Engine Torque Due to Gas Pressure," by E. S. Taylor and E. W. Morris.

[6] See Transactions of the Society of Naval Architects and Marine Engineers, Vol. 33, 1925; "Torsional Vibration in the Diesel Engine," by F. M. Lewis.

rangement of the cylinders and the arrangement of the crank throws. Almost all engines with more than four cylinders are inherently in balance, that is, at least the primary and secondary forces and couples are in balance. Of course, there are freaks such as the 90-deg. V8 with a 180-deg. crankshaft, and the 60-deg. V8 with the 90-deg. crankshaft, and so on, but the designers of these engines were aiming at special features other than balance and were willing to compromise the balance to obtain these special features. All of these designs are at present obsolete, I believe, and therefore deserve no particular attention. The two- and four-cylinder engines require special mounting arrangements that will be discussed later in this paper. The radial engine with one master rod and several articulated rods has some residual unbalance due to the difference in weight between the master rod and the articulated rods.[7] The unbalance in a single-row engine is a force unbalance and, in a twin-row engine, it is a couple unbalance. The design of engine mountings that are suited to this inherent unbalance has at times been a problem. The mountings for radial engines have by now been standardized in types that are fairly well suited to the inherent unbalance in the engine.

Line engines are practically free from inherent unbalance. The phrase "practically free from inherent unbalance" may stand some explanation. It means that the engine has no primary or secondary force or couple unbalance. Higher orders of unbalance exist in all engines but, beyond the secondary, they are small. A 6 has a sixth-order force unbalance; an 8 has an eighth-order force unbalance, and so on. These engines are commonly spoken of as being in perfect balance. They are for all practical purposes. The unbalance that they have is not only very minute, but it is of such high frequency that resonance with any normal mode of vibration would be impossible.

An engine inherently in balance can, however, be unbalanced by lack of proper control of the balance of the parts assembled on the crankshaft system. Maintaining the balance of this system is at present a serious problem for many manufacturers. It involves not only the balance of the crankshaft with its rods and pistons and the balance of the propeller assembly but the piloting of the propeller on the crankshaft so as to eliminate any eccentricity or cocking which would throw the blades out of track. Eccentricity produces static unbalance, and cocking produces dynamic unbalance. These two names probably were chosen for these two types of unbalance because the static unbalance can be determined by a static test, that is, on ways to determine at which angle the part is heaviest with respect to its axis, whereas dynamic unbalance can be determined only by whirling the parts to determine what couples exist. Radial-engine crankshafts are usually not dynamically balanced because the mass is so concentrated that no appreciable couple can exist. Line-engine crankshafts are usually dynamically unbalanced.

Propeller Balance

The discussion of propeller balance has been touched upon in the discussion of engine balance because the forces applied to the engine are those set up by the resultant of the unbalance in the whole rotating system. There are, however, some special problems peculiar to propeller balance. Dynamic balance of the propeller is, in some installations, quite necessary. The dynamic unbalance in propellers is mostly due to the center of gravity of the blades being out of track, that is, the center of gravity of one blade being forward of the center of gravity of the other. This may be due to any of three errors, or a combination of the three: (1) the out of track of the blade bores in the hub, (2) the center of gravity of the blade being off the axis of the blade, and (3) one blade bending more under thrust load than the other.

The track of the blade bores can be checked by simple inspection methods, but the location of the center of gravity of the blade requires a special set-up. It is very difficult to balance dynamically a propeller on an ordinary type of balancing machine because of the aerodynamic disturbance set up by the blades; the dynamic balance, therefore, must be obtained indirectly by making two static-balance readings. The blade is first set at its normal-pitch setting, and a reading taken of the static unbalance. Then the blade is rotated 90 deg. in the hub, and another reading is taken. The dynamic unbalance, that is, the unbalance in the plane of the propeller axis then comes into the plane of rotation and is read as a static unbalance.

Aside from mass unbalance, there are four other sets of exciting forces originating at the propeller:

(1) The couple set up by the unequal thrust loading of the blades. An analysis of this factor based on a weighted average of the thrust for each section of the blades shows that, with present tolerances on blade profile and blade synchronization with respect to each other, the couple introduced by unequal thrust loading is relatively small.

(2) The variation in thrust loading on individual blades as they pass obstructions, for example, the propeller tip passing the fuselage in bi-motor installations and the propeller tip passing the float in single-engine seaplanes. This factor is known to be of considerable magnitude, but no accurate data are available to use in calculating the forces set up.

(3) The variation in thrust loading due to difference in angle of attack in different portions of the actuator disc or disc of propeller revolution. This effect is most noticeable in climb, dive, or yaw and is a function of the airplane shape. This item requires an accurate analysis of the air flow through the propeller disc which is not available at present.

(4) With two-blade propellers, the couples set up by the variable gyroscopic moment produced when the airplane is turning about a vertical axis or pitching about a horizontal axis. Vibration from this source is most noticeable when the airplane is turning on the ground and may be extremely violent if the corresponding natural frequency of vibration of the engine on its mount is near the taxiing speed of the engine. This type of excitation is not present in three-or four-blade propellers because the gyroscopic moment is constant throughout the revolution.

Aerodynamic Excitation

Flutter and buffeting are the two main types of aerodynamic disturbance that set up vibration in aircraft structures. Flutter is a self-excited vibration resulting from an unstable loading characteristic of an airfoil. Flutter of propeller blades, although uncommon, has been found to exist in some designs using very thin airfoils. Wing flutter is undoubtedly the most serious type of flutter experienced in airplane structures, although flutter in the tail group is not so uncommon in new designs.

Although this problem is mostly outside the scope of this paper, a brief description of the causes of flutter undoubtedly will be of interest.

Flutter

Flutter is a self-excited vibration phenomenon involving the interaction of the elastic and aerodynamic forces on an airfoil. When a change in the angle of attack of an airfoil in a uniform stream of air causes the forces acting on it to shift from a condition of resisting the change to a condition

[7] See *Ingenieur* – Archiv, December, 1929; "Contribution to the Theory of Mass Balance of the Radial Motor," by P. Riekert.

of aiding the change, the fundamental requirements for the production of flutter are satisfied. According to some authorities the requirement that the system have two degrees of freedom also must be satisfied, that is, bending as well as torsion.[8,9]

The buffeting[10] of the tail groups is caused by aerodynamic disturbances set up by the structure ahead of it, particularly at low speed. Vortices form and break away from the trailing edge of the wing and the wing root, and these swirls of air flow back and strike the tail group. Disturbances set up in the slipstream by propeller-blade passages undoubtedly have an important effect on buffeting, particularly when the frequency of the blade passages coincides with the natural frequency of the formation and breaking away of these vortices. The twist of the slipstream by the propeller also has an effect on buffeting as evidenced by the fact that, in some cases, these disturbances originate on one side of the fuselage only. (The motion of the air due to twist would be upward on one side and downward on the other.)

Flutter and buffeting are problems that need a great deal of research to bring them up to the status of other aircraft-vibration problems. Designers need more empirical data for the calculation of the torsional stiffness of wings, particularly long wings of the highly tapered variety. Experimental engineers need simpler methods for determining the torsional stiffness of experimental wings.

Response

The vibration response of a system is given by the curves of vibration amplitude vs. frequency. In a mathematical study of the vibration of an engine installation there are two fundamental relationships and these two can almost be boiled down to one. They are illustrated in Fig. 1 and Fig. 2. In each figure the ordinates are non-dimensional units of vibration amplitude, and the abscissae are non-dimensional units of frequency. The ordinates are amplitudes at the specified frequency divided by a value known as static deflection. (The static deflection is the deflection which would be obtained if the maximum value of the exciting force were applied statically.) The abscissae are actual frequency divided by the natural frequency of the system. Thus resonance occurs at a frequency of unity, and all other frequencies are plotted as a percentage of resonance. Fig. 1 shows the relationship between amplitude of vibration and frequency when the amplitude of excitation is constant throughout the frequency range. The excitation from the power impulses of an engine fall into this class when the engine is used with a controllable propeller, that is, when the torque of the engine is constant throughout the speed range.

[8] See N.A.C.A. Report No. 496, 1935; "General Theory of Aerodynamic Instability and the Mechanism of Flutter," by T. Theodorsen.

[9] See N.A.C.A. Report No. 285, 1928; "A Study of Wing Flutter," by A. F. Zahm and R. M. Bear.

[10] See N.A.C.A. Technical Note No. 460, May, 1933; "Full-Scale Wind-Tunnel Research on Tail Buffeting and Wing-Fuselage Interference of a Low-Wing Monoplane," by M. J. Hood and J. A. White.

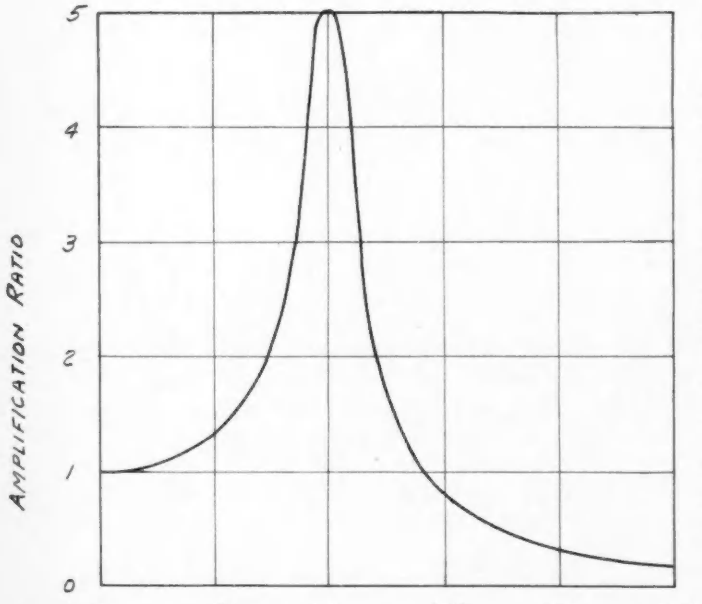


Fig. 1 – Response Curve for System with Constant Excitation

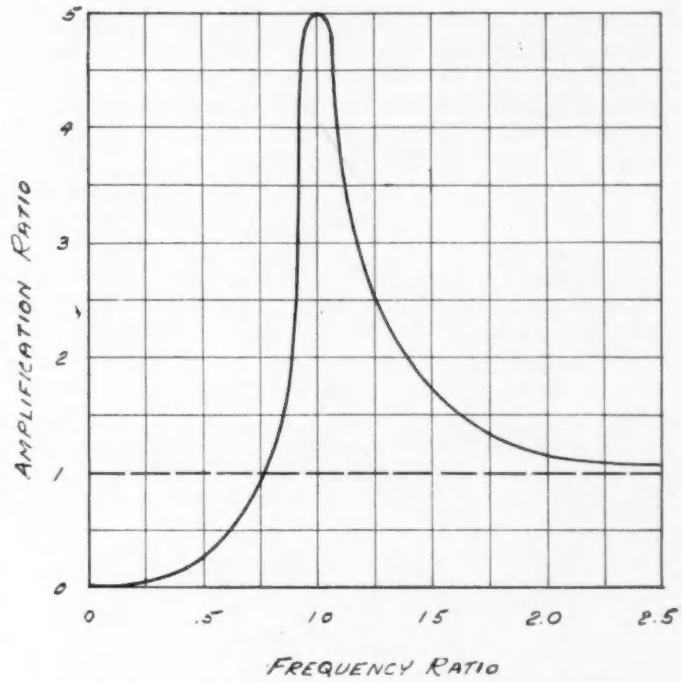


Fig. 2 – Response Curve for System with Excitation Proportional to the Speed Squared

Fig. 2 shows the relationship between amplitude of vibration and frequency when the excitation increases in proportion to the square of the frequency. The excitation from unbalanced reciprocating and rotating masses, and the excitation from the torque impulses of an engine with a fixed-pitch propeller fall into this class. It will be noted that the amplitude of vibration is zero at zero frequency. This is natural since the excitation is zero at zero frequency. The amplitude of vibration at high frequency approaches a value of unity, the static deflection. In the case of an unbalance, the static deflection turns out to be the unbalance divided by the mass of the engine, that is, $\delta_{st} = wr/W$. If the excitation is not an unbalance, it can be converted to an equivalent unbalance. This calculation can be made assuming any speed because the actual excitation and the excitation of the equivalent unbalance will be identical at all speeds.

Now that we have discussed the units of these fundamental curves, we can study their shape. It will be noted that the amplitude at resonance is several times as large as the static amplitude. The exact ratio between these two, usually referred to as the dynamic amplification, is a function of the damping in the system. The fact is that, using the customary non-dimensional damping coefficient, the dynamic amplification factor is the reciprocal of the damping coefficient. The predetermination of the dynamic amplification of airplane and engine structures is one of the most important problems before the designers of today. It is astonishing to learn how

large these factors may be and to learn what wide differences may exist in seemingly similar constructions. Take for example, the phenomenon of torsional vibration of crankshaft systems. A great deal of study has been given to this subject and the literature may be used to work out such comparisons as these:

Installation	Amplification Factor
Marine, Diesel	50 – 100
Stationary, Diesel electric	50 – 80
Automobile and truck engines	8 – 80
Aircraft engines	10 – 25
Auto pitch and bounce	As low as 1 (Using shock absorbers)

It is indeed surprising to discover that the amplitude of vibration, and therefore the vibration stress can be, in a practical installation, 100 times as great as when the load is applied statically. The preceding figures, however, are taken for systems without vibration dampers. Many of these systems immediately would break all to pieces if an attempt were made to operate them at resonance without a damper.

The range of 8 to 80 for auto and truck engines is very wide, but it can be split up into groups and the possible dynamic amplification predicted within about 10 per cent when all of the design factors are known. For large marine and stationary Diesel installations this accuracy probably can be bettered but, for aircraft engines, the 10 per cent accuracy would be very good.

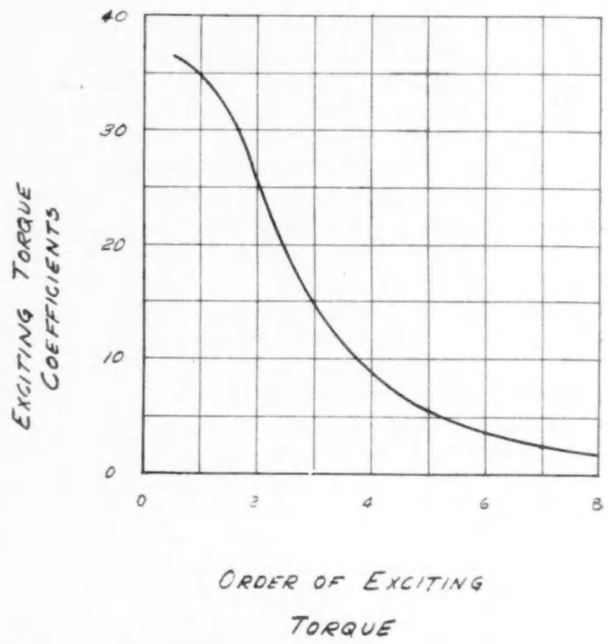


Fig. 7 – Exciting Torques for Otto Cycle Engines (Gas Torque only for Single Cylinder)

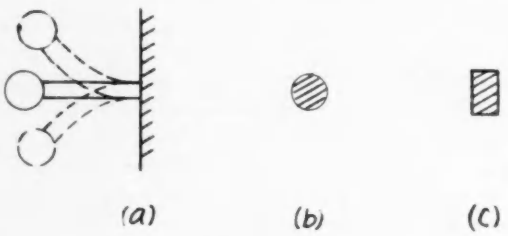


FIG. 3

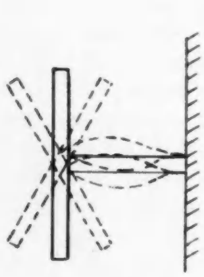


FIG. 4

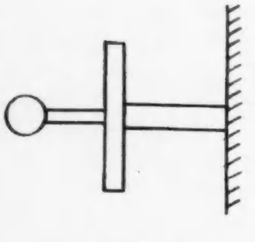


FIG. 5

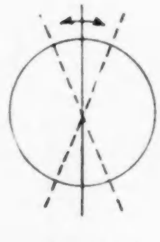


FIG. 6

Fig. 3 – Mass Concentrated at a Point and Supported at the End of a Massless Elastic Arm

Fig. 4 – Mass with an Appreciable Moment of Inertia about a Horizontal Axis

Fig. 5 – Mass of Fig. 4 with Addition of Elastic Arm with a Concentrated Mass Attached

Fig. 6 – Torsional Mode of Vibration of the Engine

The determination of these dynamic amplification factors, in some few cases, can be made analytically. In most cases, however, they must be worked out from experimental data on similar systems. For example, the dynamic amplification of a particular mode of vibration of a propeller can be determined by measuring the excitation from the crankshaft and measuring the amplitude of vibration of the propeller. With these factors available for a family of propeller-engine installations it would be possible to predict from design data what amplitudes and stresses could be expected from an installation of a new design of propeller. Studies of this nature also are being made to obtain empirical data for use in predicting the amplitude of vibration of engines on their mountings. Studies to determine similar physical constants relating to the aerodynamic disturbances around the airplane would be very helpful.

Modes of Vibration

The description of the response of vibrating systems so far has covered only the response of systems with only one resonant frequency, that is, systems with only one degree of freedom. The vibration response of an engine installation, however, includes several resonant frequencies. There are two reasons for this multiplication of trouble. The first is that the engine system can vibrate in several different ways, that is, in different modes, and the second is that there are several different sets of exciting forces set up in the installation. Some of these exciting forces produce resonance at one speed, and others produce resonance of the same type at other speeds.

The number of possible modes of vibration of a system similar to an engine installation can be explained as follows:

Assume first a mass concentrated at a point and supported at the end of a massless elastic arm as shown in Fig. 3(a). This system will have one natural frequency in the vertical plane. If the elastic arm has the same stiffness in the vertical

plane as in the horizontal plane, as indicated by the cross-section at (b), the vertical and horizontal natural frequencies will be the same. If the arm stiffnesses are different in the two directions, as indicated by the cross-section at (c), the natural frequencies will be spread apart.

Next, assume a mass that has an appreciable moment of inertia about a horizontal axis through its center of gravity, a circular disc for example. If this system is excited by an alternating couple it will have an additional natural frequency of vibration as shown in Fig. 4. Again, if the stiffness of the arm is different in the vertical and horizontal directions, there will be two additional natural frequencies for the vibrations in these two directions. The next step is to add an additional elastic arm with a concentrated mass attached as shown in Fig. 5. This arrangement will add still another natural frequency and will add two if the stiffness is different in the vertical and horizontal directions. The next step would be to make the additional mass a disc and add two more possible modes of vibration.

To bring this explanation down to an actual installation, we find that all of these modes actually exist. The engine,

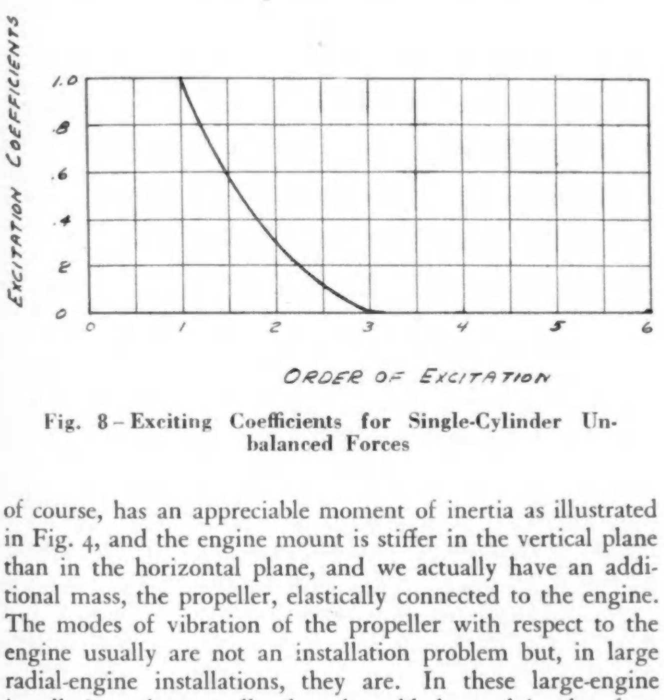


Fig. 8 – Exciting Coefficients for Single-Cylinder Unbalanced Forces

of course, has an appreciable moment of inertia as illustrated in Fig. 4, and the engine mount is stiffer in the vertical plane than in the horizontal plane, and we actually have an additional mass, the propeller, elastically connected to the engine. The modes of vibration of the propeller with respect to the engine usually are not an installation problem but, in large radial-engine installations, they are. In these large-engine installations the propeller has three blades and is, therefore, essentially a disc from a vibration standpoint. The engines are geared and the propeller shaft and crankcase stiffnesses are identical in the vertical and horizontal directions. In smaller radial-engine installations where the propeller has two blades and is mounted on the crankshaft which is stiffer in the plane of the throw than in a plane perpendicular to it, the two natural frequencies involving shaft bending will be spread farther apart.

The modes of vibration that are most troublesome in the small installations are the four illustrated in Figs. 3 and 4 plus a torsional mode of vibration of the engine shown in Fig. 6. The modes of vibration that are most troublesome in large installations are these five just mentioned plus the two added by the propeller. The spread between these last two is the result of different vertical and horizontal stiffnesses in the engine mount.

Harmonics of Excitation

Single frequencies of excitation, that is, pure sine-wave excitation, does not appear very often. For example, middle C on the scientific scale is 256 cycles per sec. If middle C on a piano were tuned to 256 cycles per sec., it also would produce notes with frequencies of 512, 1024, 1536, and so on. Similarly, a violin always produces not only the fundamental note being played but also a large number of harmonics or overtones. The difference in quality of the same notes from different musical instruments is due to the difference in the number and proportions of the overtones.

The torque impulses of an internal-combustion engine are not pure sine waves either. Thus, when the engine is firing at 150 cycles per sec., there are also torque impulses at 300, 450, 600, and so on, cycles per sec. and, if propeller-blade tips pass the fuselage at a frequency of 60 cycles per sec., there also will be large excitation impulses at 120, 180, and so on, cycles per sec. The effect of these harmonics is to multiply the number of times a given mode of vibration appears. For example, if the natural frequency of vibration of an engine in the vertical direction as shown in Fig. 3 is 2100 cycles per min., the primary unbalance will produce resonance at 2100 r.p.m. and the secondary unbalance will produce resonance at 1050 r.p.m. The vibration frequency will be the same in both cases because the frequency of the secondary unbalance at 1050 r.p.m. is 2100 cycles per min.

The curve shown in Fig. 7 shows the relative magnitude of the harmonics of the torque curve. This relationship is fairly typical. The higher orders of excitation usually decrease gradually, and the curve becomes asymptotic to the zero line. In some cases, the curve drops quite suddenly as in Figs. 8 and 9, showing the magnitude of the orders of inertia force and inertia torque unbalance for a single-cylinder engine.

Another factor that determines the importance of a vibration period is its location in the speed range of the engine. The example given two paragraphs earlier of a first-order vibration at 2100 r.p.m. and a second-order vibration at 1050 r.p.m., is rather typical. If 2100 r.p.m. is an operating speed, that period is more objectionable than 1050 r.p.m., although 1050 r.p.m. is close to the usual warm-up speed of 1000 r.p.m. and is, therefore, more objectionable than 1300 r.p.m. or 800 r.p.m. Again, suppose we have a nine-cylinder radial engine with a 4½ order torsional vibration at 1750 r.p.m. The 9th order vibration will occur at 875 r.p.m. and the 13½ order will occur at 583 r.p.m. The 9th and 13½ orders will be negligible for three reasons; the harmonic torques are small (see Fig. 7), the engine torque is small, and the periods occur outside of the operating-speed range. Occasionally, however, this condition is reversed and the 4½ order is at 5000 r.p.m., with the 9th at 2500 r.p.m., and the 13½ at 1666 r.p.m. In this case, if the 9th order is just above the operating speed and the 13½ order is just below it, the

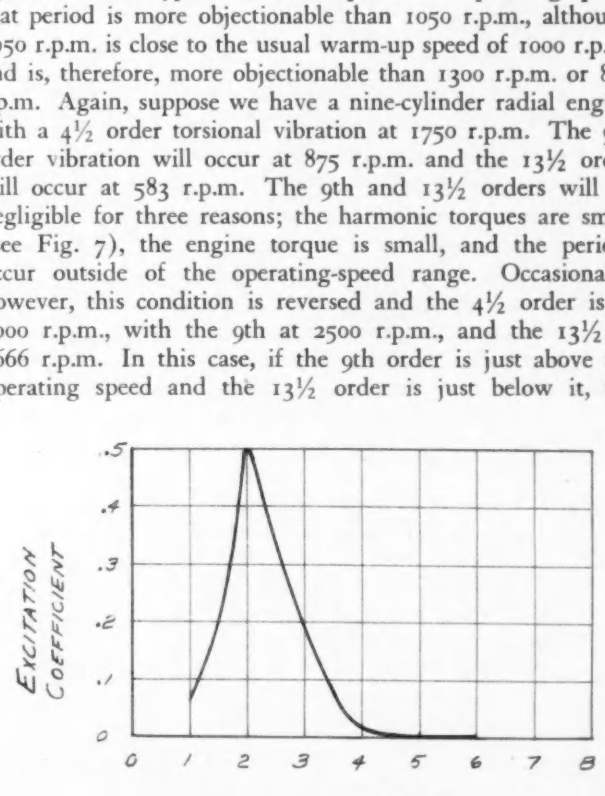


Fig. 9 – Exciting Coefficients for Single-Cylinder Inertia Torque

former will be serious and the latter insignificant because of the ratio of the exciting torques (see Fig. 7).

One other factor of prime importance is the location of the exciting force with respect to the nodes in the system. Going back to Fig. 3, it will be obvious that an exciting force at the concentrated mass will be most effective in producing vibration. The mode shown in Fig. 4 will be produced most effectively by an exciting couple acting about the center of gravity of the disc. Conversely an exciting couple acting on the concentrated mass of Fig. 3 or an exciting force acting on the disc of Fig. 4 will produce no resonant vibration in these respective modes except to a minor extent due to the angular motion involved in the linear motion of the masses.

In an actual engine installation the nodes ordinarily do not locate themselves in the plane of the excitation unless the mounting is specifically designed to accomplish this arrangement. The lowest two modes, horizontal and vertical, indicated in Fig. 3, usually have a node behind the engine, in the vicinity of the fire wall. The modes indicated in Fig. 4 may have the node either forward, aft, or in the plane of the centerline of the crank throw. It will be obvious that the first mode will be excited most easily by static unbalance in the installation and that the static unbalance in the propeller will be more effective in producing vibration than static unbalance in the engine. In the second mode if the node is on the centerline of the crank throw the static unbalance in the engine will be entirely ineffective in producing vibration but static unbalance in the propeller will still be very effective. In this mode dynamic unbalance in the engine (as in a double-row radial) will be very effective in producing vibration as also will the dynamic unbalance in the propeller.

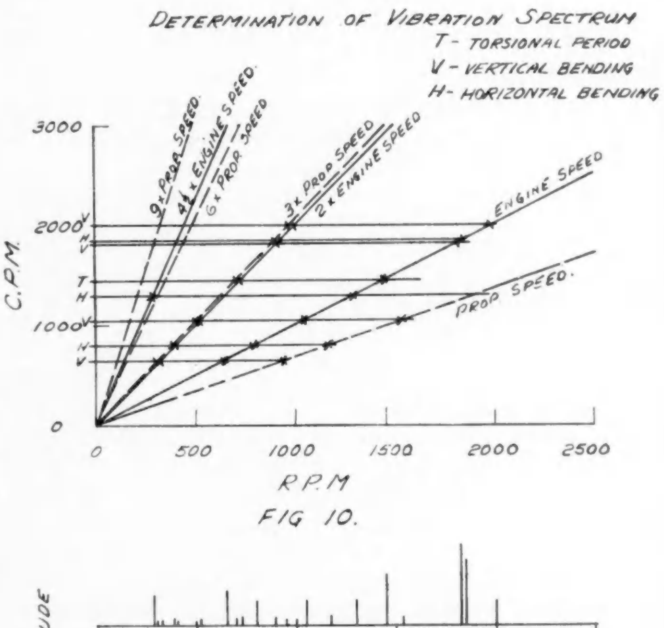
In the design and development of an engine mount all of the previously mentioned factors should be kept in mind. The best mount for a given installation is usually a compromise on several different features. The usual method of developing a mount for a new installation is to try all of the possible combinations of rubber bushings and steel structures that can be thought of. By this cut-and-try method nodes are moved forward and backward, and natural frequencies are moved up and down in the speed range. The engineer does not know what is going on, but he chooses the best combination that he can find. When he is through with the job, he may not have the best combination, but he has one that will work.

A more scientific method of developing an engine mount is to determine from ground studies, using artificial means of excitation, what and where all of the modes of vibration are. The first step is then to determine which modes are going to give the most trouble with the engine and propeller balance that must be expected using normal production tolerances. In case of a new design, this determination should be made on an engine mock-up before fabrication of the first plane.

At this point the vibration spectrum should be made to show the location of each mode of vibration with respect to each of the separate sets of exciting forces. As an example a vibration spectrum is shown in Fig. 10. The abscissae are engine r.p.m. The ordinates are the exciting frequencies set up by the engine installation. The natural frequencies of the various modes of vibration of the engine installation are located by horizontal lines. These data are for a bi-motor installation of a geared nine-cylinder radial engine with a 16:11 gear ratio and a three-blade propeller. From this engine and propeller combination there will be four separate sets of exciting forces: the unbalanced forces of the radial engine at crankshaft speed and twice crankshaft speed, the torque impulses of the engine at 1, 4½, and 9 times crankshaft speed, the unbalanced forces from the propeller at propeller speed and the propeller-tip interference forces at 3, 6, and 9 times propeller speed. All of these exciting forces can be plotted as straight lines starting at the origin and having a slope equal to the number of cycles of excitation per revolution of the crankshaft.

Wherever the frequency of the exciting forces is equal to the natural frequency of the vibration of the system, resonance will occur. As pointed out in earlier paragraphs, if the exciting force is near a node or, if an exciting couple is near an anti-node, the exciting forces will be ineffective in producing resonant vibration. Of course, there will be no response from the engine torsional exciting forces of the engine except the torsional vibration of the engine unless there is some unusual coupling between this mode and other modes. For this reason the torsional mode is shown as a dotted line on the chart and the torsional excitations are shown as dotted lines. The torsional excitation of 1 cycle per engine revolution (this is due to non-uniformity of firing impulses) is a maximum under warm-up conditions because of the poor fuel distribution at low speeds with a cold engine. For this reason it is highly desirable to keep this resonant period away from the warm-up speed. With the engine windmilling at this speed, that is, with the throttle closed and the engine operating at 1000 r.p.m., the 1 cycle per revolution torsional excitation should be very small.

The resonance conditions that should be investigated are indicated in Fig. 10 by X's and are shown on the chart in Fig. 11 as lines of a spectrum. This type of spectrum is made up as a chart of vibration intensity against engine r.p.m., the



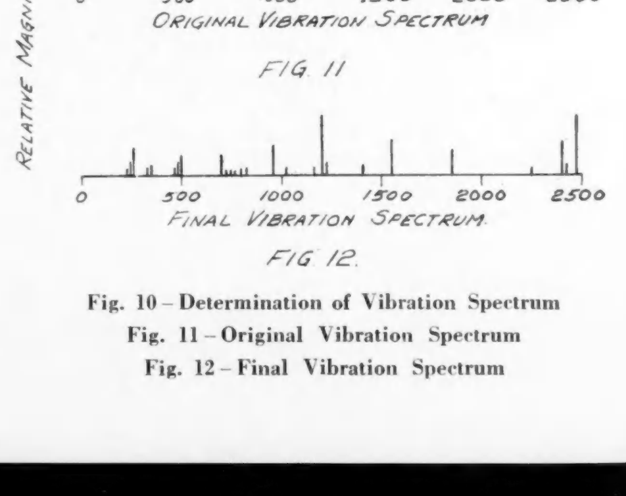


Fig. 10 – Determination of Vibration Spectrum

Fig. 11 – Original Vibration Spectrum

Fig. 12 – Final Vibration Spectrum

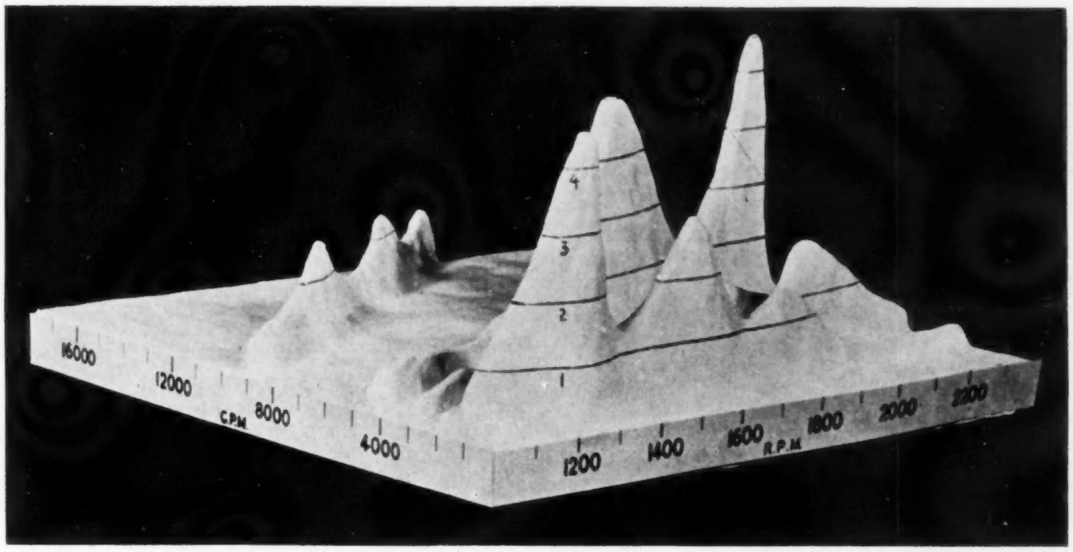


Fig. 13 – Three-Dimensional Chart Showing the Vibration Characteristics of an Engine Installation

intensity of the vibration being indicated by the length of the line. It will be noted from this chart that several modes of vibration exist in the operating-speed range. These modes either must be shifted to locations outside of the operating-speed range or must be reduced in magnitude by some means or other. The best solution is to move them outside of the operating-speed range if this is possible. This particular engine installation was changed to give the vibration spectrum shown in Fig. 12 with all of the most objectionable periods outside of the operating-speed limit. The detailed changes in the mount that were required to change the spectrum from that shown in Fig. 11 to that shown in Fig. 12 include several of the tricks suggested in previous paragraphs; for example, shifting periods out of the operating-speed range, reducing the effectiveness of unbalance by shifting nodes toward the plane of unbalance, and moving the higher damping material (rubber) toward the nodes. Introducing additional flexibility in the form of rubber into the system also reduced the transmission of noise and high-frequency vibration to the cabin of the airplane, but this reduction does not show up on the vibration spectrum.

The lines in the spectrum indicate the frequency at which resonance occurs but the amplitude of vibration is large over a frequency band quite appreciable in width. The curves shown in Fig. 1 and Fig. 2 indicate the width of this band as a percentage of the resonant frequency. When all of these resonance curves are added together, there will be, of course, considerable overlapping and there will be an appreciable amplitude of vibration at all frequencies.

The presentation in a single diagram of the curves of vibration amplitude vs. r.p.m. is a rather difficult problem. It requires three dimensions and must therefore be a solid. A three-dimensional chart of this type is shown in Fig. 13. The contour lines show the vibration-amplitude scale. It will be noted that there are several ridges that slope away from the r.p.m. scale. These ridges correspond to the diagonal lines in Fig. 10. The peaks in the chart are resonant points, that is, where the excitation is equal to the natural frequency of some mode of vibration. This three-dimensional chart shows the vibration characteristics of an outboard-engine installation of a tri-motor airplane.

Installation vs. Engine and Propeller

It is sometimes difficult to distinguish between vibration periods that are strictly engine and propeller vibrations and vibration periods that are to be called installation vibrations. The line is drawn, for purposes of discussion in this paper, at the point where the stiffness of the mount begins to affect the frequency and amplitude of the resonant vibrations.

In a small-engine installation (for example, a 250-hp. nine-cylinder radial) in which the mounting is low enough so that the four periods indicated in Figs. 3 and 4 are below the operating-speed range, there are no engine or propeller periods that can be affected measurably by the engine mount. The engine-propeller installation will ride as though on cushions on the airplane. There will, of course, be propeller and crankshaft vibration periods, but the pilot will not hear, see, or feel them. Their existence will not be suspected unless they are measured by special equipment or some indication of their presence is obtained from inspection of the parts.

In a large-engine installation the propeller is much heavier in comparison with the engine weight, and the bending stiffness of the propeller shaft is lower in comparison with the propeller weight. As a result the additional periods indicated by Fig. 5 are in the vicinity of the operating-speed range for large engines. These periods are in the range of 7500 to 10,000 cycles per min. for 250-hp. engines and are 1800 to 3000 cycles per min. for 1000-hp. radial engines. In the latter case the natural frequency may be shifted by as much as 1800 to 2500 cycles per min. by simple changes in the mountings.

The natural tendency, however, is toward more flexible mountings that have most of the resonance periods grouped together below 1000 r.p.m. As soon as manufacturers have mastered the difficulties of these mounts, even the propeller periods of large engines will not be an installation problem, that is, they will not be affected by changes in engine mounts.

Instrumentation

The instrumentation used in making the vibration studies of aircraft-engine resonance may be divided into three groups: exciters, pickups, and recorders.

Exciters are mechanisms used to produce artificial excitation. They can be mounted at various points on an engine to produce forces, couples, or torques as desired. The most convenient type of exciter is one which can be mounted on the propeller nut and operated at variable speeds of 500 to 5000 r.p.m. An exciter of this type is shown in Fig. 14. The Air Corps at Wright Field and Wright Aeronautical have been using exciters of this type for some time. In making vibration studies of engines and propellers, it is highly de-

sirable to have an exciter that can be operated up to 20,000 or 30,000 r.p.m. One of these high-speed exciters is shown in Fig. 15. The most convenient set-up is an exciter of this sort with a Transitorque electric-motor drive. This drive is a constant-speed motor with an infinitely variable transmission. A motor of this type is extremely stable as regards speed characteristics, which feature is extremely important in studying resonant vibrations in which the power consumption of the exciter is extremely variable with speed. The propeller-nut position for the excitation is very convenient because, from this point, the effects of both dynamic and static unbalance in the crankshaft and propeller can be duplicated. The driving torque to the exciter is taken through a rubber-hose coupling so that the installation will be free to vibrate. The high-speed exciter must be driven through a step-up gear. In the high-speed exciter shown in Fig. 15, the inner race of the ball bearing turns, and the outer race is stationary. This construction appreciably reduces the peripheral speed of the balls. This high-speed exciter was designed for continuous duty in fatigue testing. That factor accounts for its rugged construction.

The design specifications for an airplane vibration pickup always include one important requirement. It must be flyable. The last word in airplane testing must be obtained in the air. This means that the instrument must be remote recording. For this reason electrical means for recording airplane vibrations seem to have been accepted unanimously. The instruments in use are made of a permanent magnet and a moving coil. The author has been using for some time a vibration pickup built by the Westinghouse Research Laboratories.[11] This unit, shown in Fig. 16, has been modified to lower its natural frequency for use in engine-installation work where measurements must be taken at speeds down to 1000 r.p.m. and lower. The Navy has been sponsoring an instrument-development program at the Massachusetts Institute of Technology recently in which a pickup of similar design has been developed.

The output from these moving-coil pickups is proportional to the relative velocity between the coil and the magnetic field. The pickup is designed as a seismographic instrument with the magnet floating in space and the coil attached to the vibrating body. Thus the amplitude of the curve of the electrical potential at the terminals of the moving coil is proportional to the amplitude of vibration at a given frequency. However, as the frequency of the vibration changes, the output of the pickup changes in a direct proportion. Professor Draper at Massachusetts Institute of Technology uses an integrator in conjunction with the pickup to convert the velocity reading to displacement. However, most vibration studies are made by comparisons of "before" and "after," so it makes little difference whether the readings are in velocities or displacement. If the data must be converted into

[11] See *Journal of the Acoustical Society of America*, July, 1933; "A New Portable Meter for Noise Measurement," by W. A. Osbon and K. A. Oplinger.

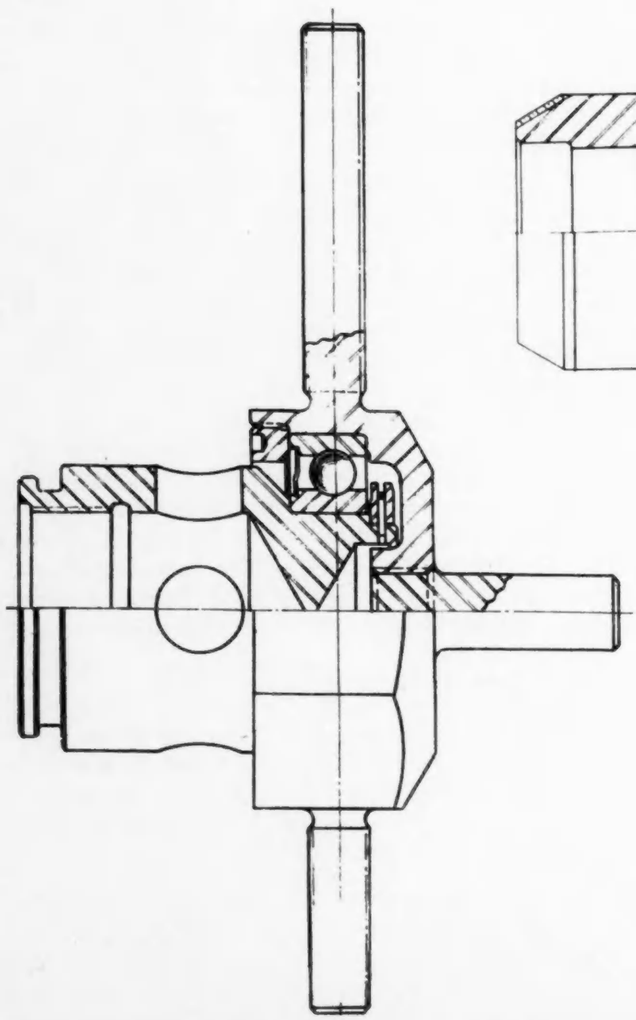


Fig. 14 – Cross-Section of Low-Speed Exciter

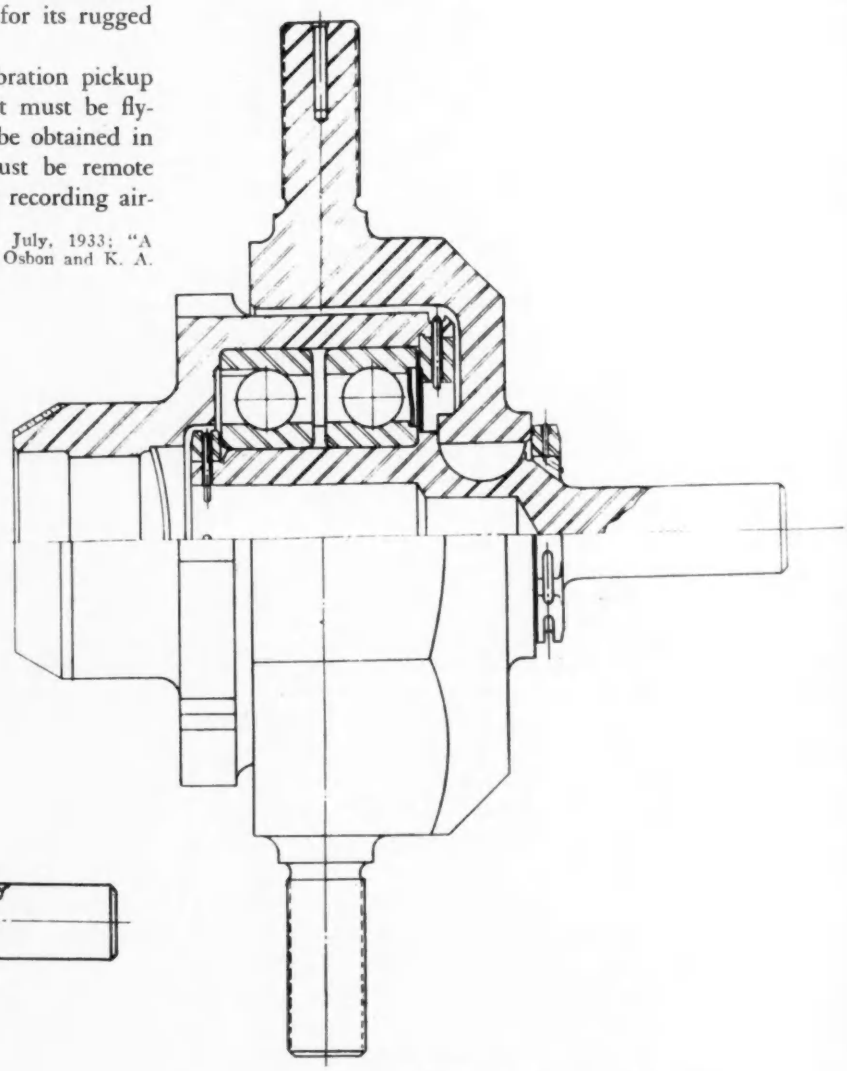


Fig. 15 – Cross-Section of High-Speed Exciter

inches of displacement, the readings must be divided by a calibration constant and, in this operation, the conversion from velocity can be made by dividing also by frequency.

There are two methods of obtaining the data from these pickups. The most common method is to make photographic records with an oscillograph. Another method involves the use of an harmonic analyzer, an instrument which responds to only one frequency at a time. The analyzer can be used to read the amplitude separately of each frequency present in a system. Records are made by the operator copying down the readings of a meter. A photograph of the Westinghouse amplifier and analyzer used by the author is shown in Fig. 17.

There are several advantages of the analyzer over the oscillograph, the most important of which are as follows:

(1) The analyzer results are immediately available. The operator can plot curves of amplitude vs. r.p.m. as he takes the data and can fill in points to be sure that he has the peak values. When the test flight is finished, he can prescribe immediately what changes to make before the next test. With an oscillograph, the operator does not know what the results are until after the films are developed and the readings obtained then are plotted. Thus considerable time is lost and, occasionally, tests must be re-run because of insufficient data.

(2) Oscillograph records are often quite difficult to interpret because of the presence of several different frequencies on the same record. The analyzer overcomes the difficulty by reading each frequency separately. There are occasions, however, where it is desirable to determine the phase relationship between various frequencies of a composite vibration. On these occasions, which are rare, oscillograph records should be made.

(3) The oscillograph record is only a small sample of a continuously varying phenomenon, whereas the analyzer gives a continuous reading on a meter.

Fig. 16 – Low-Frequency Pickups (*left*) for Use in Vertical Direction (*below*) for Use in Horizontal Direction

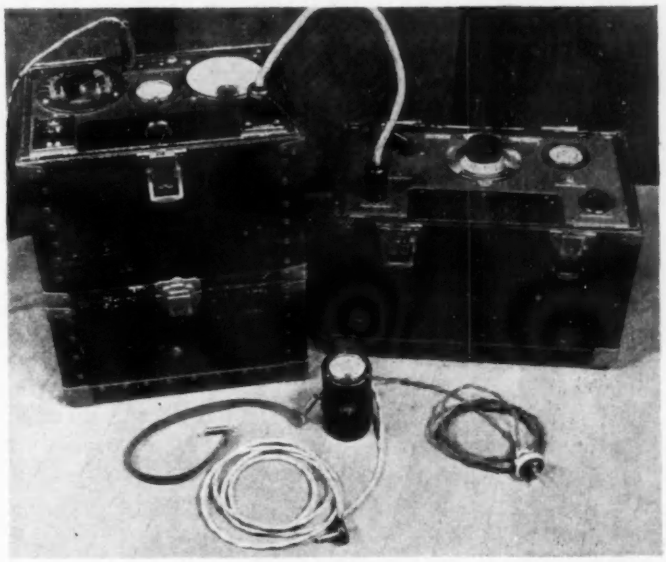


Fig. 17 – Amplifier and Analyzer – A Calibrator is Shown in the Foreground

Suppression of Vibration

In airplane installations the use of gadgets to act as vibration dampers (hydraulic dampers, brake lining, and so on) is not looked upon with favor. If an installation is giving trouble and a remedy is required that will not upset the production set-up, a gadget may be justified to pull out of a hole temporarily. But the proper method of suppressing resonant vibration is to design so that it comes out of the operating speed range or design to make the excitation ineffective in producing vibration at resonance.

An engine mount really has two functions as far as vibration is concerned:

(1) It must insulate the vibrations inherent in the installation from the airplane structure. This insulation is done most successfully by using lots of rubber. Just how the rubber is shaped makes little difference as long as it is not pre-loaded too heavily and has as much distance between the metal of the engine and the metal of the airplane structure as possible.

(2) The second function of the mount is to place all objectionable resonance frequencies of the engine with respect to the airplane out of the operating-speed range, preferably below the operating-speed range. This function requires a fairly flexible mount. The resonant frequencies must be sandwiched into the range below the operating-speed range so that the engine will not bounce around too much at idling and warm-up speeds.

A scheme that is receiving some attention at present is that of pre-loading the rubber units in the engine mount against the engine torque. The mount will then be relatively stiff at idling and warm-up speeds and, when the throttle is opened for take-off, the engine torque twists the engine away from the pre-loading stops, and the mount is very flexible. The deflection of the engine under cruising torque must be great enough so that the mount will not strike the stops too violently with the normal amount of variable distribution and occasional poor spark-plugs. There also would be a part-throttle gliding condition for this mount which would have to be avoided.

Future Development

The changes involved in the improvement of the vibration characteristics of airplanes are quite definite as far as vibra-

tions originating from excitations within the engine are concerned. Before such improvement can be obtained by reducing the aerodynamic disturbances, some fundamental data on the dimensions of the disturbances must be obtained.

Engine mounts will be more flexible and, with improved technique, the location of each of the resonant periods will be determined by design rather than by chance.

Improvement in engine and propeller balance, in some cases, may be necessary but, for small engines, a satisfactory installation undoubtedly can be obtained for each airplane by proper design of the engine mounts. In large engines where unbalance produces resonant vibrations within the engine-propeller unit, an improvement in engine balance will be highly desirable.

An increase in the number of cylinders beyond nine will have no influence on the amount of vibration transmitted through the mounting if the proper mounting is used in each case. An increase in the number of cylinders will, however, simplify the problems of smoothing out the vibrations at idling, warm-up, and windmilling speeds.

With the reduction of vibration transmitted to the cabin from the engines, the propeller noise has become much more prominent. It seems that some reduction of propeller noise in the cabins must be made in the near future. This prediction is based upon the fact that most of the present noise comes from aerodynamic disturbances around the propellers, fuselage, and wing, and we are sure that considerable improvement will be made. As to whether the improvement must be made by improved cabin insulation or whether some gain can be obtained by reducing the disturbance will depend on the characteristics of the disturbance, which characteristics are still unknown.

Discussion

Stresses Importance of Dampers and Connecting-Rod Structure

– Earl V. Farrar

Test Engineer, Wright Aeronautical Corp.

IN discussing Mr. Tyler's paper, the writer would first like to pay compliment to his able discussion of a subject that apparently has been little understood in the past, or perhaps, has not received its share of attention due to the overshadowing importance of such other problems as getting more miles per hour from the airplane, carrying more load per horsepower, keeping the engine cool, getting more miles per gallon, and so on. Moreover, the engine mount is that structure which at present bridges the gap between the product of two separate and distinct manufacturers, neither of which have been able to give it the attention which it deserves. Large-scale production of transport planes, in which a comfort-conscious public travels, is rapidly focusing attention of both engine and airplane manufacturers on the powerplant installation and its vibration problems. It is hoped that Mr. Tyler's excellent paper will clarify the vibration problems involved in aircraft-engine installations and pave the way for a great deal more cooperative work between engine and airplane manufacturers on this important subject.

However, the writer somewhat disagrees with Mr. Tyler in his view that dampers may not be used properly to suppress resonant vibration. In the section headed "Response" it is inferred, but I believe not directly stated, that at resonance the amplitude of a vibrating mass becomes infinite if no dampening whatever is present, regardless of the magnitude of the exciting force. This being the case, no engineering structure could exist under the influence of a cyclic disturbance with frequency equal to the natural frequency of the structure. It is obvious then that damping is one of the most important factors in preserving structures of all kinds. The writer agrees that the exciting forces should be reduced to a minimum and that the transmission of forces from engine to airplane due to the inevitable relative motion of the engine and airplane sections should be minimized by "soft mounts". It also is believed that dampers, particularly of the dynamic type, may be an important aid in subduing the effects of resonance between exciting forces and the natural frequency of the engine-supporting structure.

Mr. Tyler also mentions engine unbalance due to the connecting-rod structure in common use in radial-type engines. The writer hesitates to condemn such a useful and reliable mechanism as this type of rod system has shown itself to be, but it is certainly responsible for a great deal of the "roughness" experienced with large radial engines. In single-row radial engines in the 800-hp. class, there is a rotating unbalanced force moving at twice the angular velocity of the crank of approximately 2000 lb. at rated engine speed. In two-row radials, there is a correspondingly large unbalanced secondary couple. As a characteristic of the master and articulated-rod system, a once-a-revolution inertia torque variation equal to approximately 10 per cent of the normal gas torque variation also exists. This once-a-revolution torque variation is increased to 25 per cent of the normal 4½ order torque variation (in a nine-cylinder engine) by timing and valve-event variation inherent in the conventional rod system. The obvious method of eliminating these exciting forces is to find a connecting-rod system that will impart equal motion to all the pistons of a radial engine. What this mechanism should look like, the writer is not prepared to say.

High Output in Aircraft Engines

(*Continued from page* 231)

effective pressure may be attained without excessive gear ratios in an engine of high crankshaft speed. Thus the problem of shock reduction and gear design becomes greatly simplified.

Accessories Drive

The trend in accessory design is toward higher speeds; and, as in the case of the supercharger drive, the higher the crankshaft speed, the lower the gear step-up ratio required.

Propeller Reduction Gear

In the design of this unit, increasing the speed of an engine of high mean effective pressure definitely involves an increase in size and weight, at least with the types of gears in use today. Since there is a distinct trend toward lower propeller speeds, it is possible that the near future will see the evolution of types of gears quite different from those now in use, in order to accommodate the higher gear ratios.

Conclusion

In conclusion, it may be well to call attention to the fact that the recent outstanding advances in fuel for spark-ignition engines have made possible specific outputs and fuel economies which the engine designer as yet has not been able to use to advantage. The purpose of this paper will be served if it will promote developments permitting the full utilization of such fuels.

One of these developments is that of improved pressure-measuring technique. Lack of such technique may well be expected to involve considerable cylinder damage and produce a state of "maximum-pressure consciousness" among engine designers.

The development of pressure inhibitors (as distinct from detonation inhibitors) must be pursued, which inhibitors should be as effective as water but not require separate injection means. The cyclic variation in maximum pressure should be reduced, and we suggest as a means to that end that spark timing with spark-plugs having electrodes with various degrees of electron emissivity be investigated. The thought occurs that, with the increase in mean effective pressures permissible with proposed fuels, an inevitable increase in engine speed will result and the problem of visualizing extremely high output engines becomes simpler as we accept the high-speed engine and properly weigh the design features as affected by pressures and speeds.

In the preparation of this paper we have continuously drawn from experience gained during work done for the U. S. Army Air Corps, and it is with gratitude that we acknowledge the Materiel Division's assistance and permission to present these speculations.

Compression-Ignition Engine Performance at Altitude

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Assistant Mechanical Engineer *Junior Mechanical Engineer*
National Advisory Committee for Aeronautics

ENGINE-TEST results are presented for simulated altitude conditions using a displacer-piston combustion-chamber on a 5-in. by 7-in. single-cylinder compression-ignition engine operating at 2000 r.p.m.

Comparison between maximum performance at altitude of the compression-ignition engine and a carburetor engine showed that the compression-ignition engine had a slight power advantage for the same conditions of inlet air. However, if the carburetor air is heated to prevent icing, the compression-ignition engine inducting the colder and more dense air of altitude will have a decided advantage over the carburetor engine.

Analysis of the results for which the inlet-air temperature and pressure were varied independently indicates that maximum engine performance cannot be corrected reliably either on an inlet-air-density or weight-of-air-charge basis. Maximum engine power increases with inlet-air pressure and decreases with temperature very nearly as straight lines. Correction factors are suggested accordingly.

THE effect of altitude on the performance of carburetor engines has been investigated thoroughly by numerous agencies, but very little actual experimenting has been done on the effect of altitude on the performance of a compression-ignition engine. The purpose of this paper is to present the results of tests that have been made at the Langley Field laboratories of the National Advisory Committee for Aeronautics on a compression-ignition engine for various combinations of inlet-air temperature and pressure to simulate altitude conditions.

In order to make the most exacting altitude tests on any type of engine, an altitude chamber similar to the one at the National Bureau of Standards[1] is desirable; however, lacking such equipment, reliable results can be obtained by the methods described in this paper. The general test procedure used in this investigation may be called an "approximate" method. Results obtained by this approximate method have been compared directly with results obtained by the National Bureau of Standards in an altitude chamber.[2] Tests were made in the altitude chamber and, immediately thereafter, they were repeated using the approximate method. The discrepancy between the results obtained by the two methods was so small as to be within the limits of experimental accuracy. The approximate method of making altitude tests has been used for some time by the Naval Aircraft Factory for testing carburetor engines.[3]

When considering the effect of altitude on engine performance, it should be remembered that all changes in altitude are fundamentally changes in air density as a result of changes in the temperature and pressure. These fundamental changes are present in one form or another regardless of supercharging, and the effect on engine performance is vitally important. The scope of these tests accordingly was broadened to include the effect of a wide range of temperatures and pressures of the inlet air. Correcting compression-ignition engine performance by standard carburetor-engine methods has been found to be incorrect and a method is suggested that, for the engine under test, gave results exceptionally close to the actual test results.

The analysis of this series of engine-performance tests will be extended and incorporated into an N.A.C.A. report to be published at a later date.

Test Engine

The development of the displacer-piston combustion-chamber and fuel-spray arrangement used in these tests (see Fig. 1) has been described completely in earlier Committee publications.[4, 5] The more important parts of the test unit and some test conditions were as follows:

Engine	Single-cylinder, four-stroke cycle, 5-in. bore by 7-in. stroke (137.45 cu. in. displacement).
Engine speed	2000 r.p.m.
Compression ratio	14.5.

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 12, 1937.]

[1] See A.S.M.E. Transactions, 1932, AER-54-8, pp. 53-60; "Altitude Laboratory Tests of Aircraft Engines," by H. K. Cummings and E. A. Garlock.

[2] See N.A.C.A. Technical Note No. 210, 1924; "The Testing of Aviation Engines under Approximate Altitude Conditions," by R. N. Du Bois.

[3] See Report of Bureau of Aeronautics, Navy Department, Serial No. AEL-346, Naval Aircraft Factory, 1931; "Test of Refrigerator and Dehumidifying Equipment," by P. M. Sartell.

[4] See N.A.C.A. Technical Note No. 518, February, 1935; "Performance Tests of a Single-Cylinder Compression-Ignition Engine with a Displacer Piston," by C. S. Moore and H. H. Foster.

[5] See N.A.C.A. Technical Note No. 569, May, 1936; "Boosted Performance of a Compression-Ignition Engine with a Displacer Piston," by Charles S. Moore and H. H. Foster.

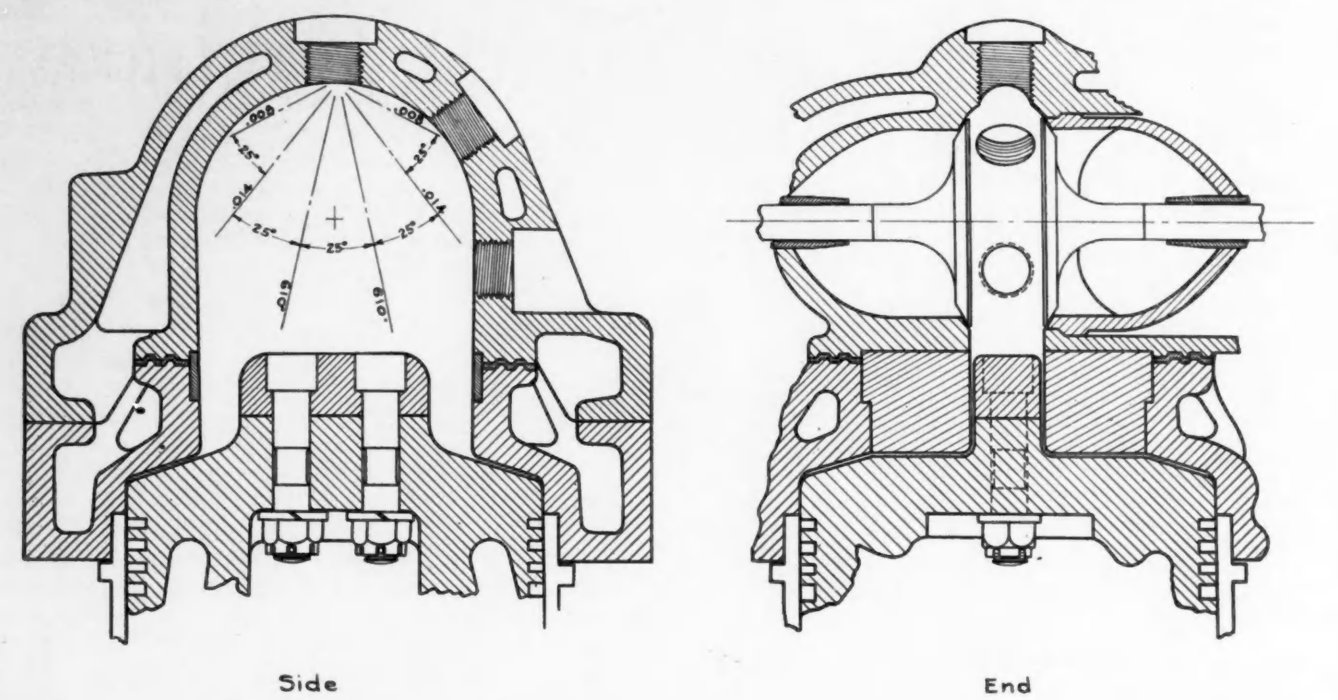


Fig. 1 – Combustion-Chamber and Fuel-Spray Arrangement

Valve timing	I. O. 27 deg. before top-center.
	E.O. 66 deg. before bottom-center.
	I. C. 28 deg. after bottom-center.
	E. C. 41 deg. after top-center.
Fuel	Auto Diesel fuel, 0.847 specific gravity, 41 sec. Saybolt Universal viscosity at 80 deg. fahr.; Cetane No. 66.
Fuel-injection pump	N.A.C.A. cam-operated, constant-stroke type.
Fuel-injection valve	N.A.C.A. automatic, spring-loaded to 3500 lb. per sq. in. opening pressure. (Injection period 25 crank deg. at 3.56×10^{-4} lb. per cycle.)
Power measurement and absorption	Electric dynamometer unit.
Blowers	Inlet – 4-in. Roots type, separately driven.
	Exhaust – 8-in. Roots type, separately driven.
Operating temperatures	Water (out), 170 deg. fahr.
	Lubricating oil (out), 175 deg. fahr.
Air and fuel-consumption measurements	Synchronized electrically operated stop watches and revolution counters.
Maximum cylinder-pressure indicator	Trapped-pressure type.

Air-Conditioning Equipment

A photograph of the equipment assembled to simulate altitude conditions is reproduced in Fig. 2. In order to show the operation of the equipment more clearly, a diagrammatic representation of the apparatus is shown in Fig. 3. By means of the gate valve *C* shown in the air line to the inlet surge tank, any reduction in inlet pressure could be maintained at the engine inlet, the pressure being indicated by a mercury manometer *E* connected at a point about 18 in. from the intake port. The Roots blower *B* connected to the exhaust tank *D* was used to evacuate the tank to correspond to the pressure of the altitude that was being simulated.

The chief difficulty in making altitude tests is, of course, the problem of reproducing the corresponding air temperature. For these tests the simple and economical air-cooling apparatus shown in Fig. 4 was designed and constructed. Because of the necessity for taking care of the moisture as it was condensed out of the air, the cooling of the air was divided into two stages. In the first stage a 50-50 solution of Prestone and water was used as the coolant. The 50-gal. tank of the coolant *G* was charged directly with solid CO_2 until the temperature was reduced to approximately -20 deg. fahr. The rate of charging was controlled to hold this temperature during the operation of the system. No difficulty was experienced in this respect and whole 40-lb. cakes of CO_2 were put into the tank.

In the second stage kerosene was used as the coolant because the temperature could be lowered without a decided increase in viscosity. As before, 40-lb. cakes of solid CO_2 were placed into the 50-gal. tank of kerosene *O*, and the temperature was maintained at -35 to -40 deg. fahr.

Inlet air for the engine was brought into the first-stage cooler for dehumidifying and cooling. This cooler *E* consisted of a radiator having 60.88 sq. ft. of cooling area with 27.72 sq. in. of area for the air flow. The rate of flow for the engine operating speed of 2000 r.p.m. was approximately 70 cu. ft. per min. at sea-level pressure. Circulation of the cold Prestone-water solution through the tubes of the radiator cooled the first-stage cooler. A secondary de-icing circulation was maintained by spraying the outside of the tubes with the same solution. This second circuit is called "defrosting" spray and was necessary to prevent the radiator from becoming clogged with ice. The air and defrosting spray passed on to the snow box *L* where the entrained solution, which had taken up the

moisture from the air, was settled out. At a temperature of about —5 deg. fahr. the cold dry air went to two kerosene coolers in parallel arrangement where it was further cooled to —30 deg. fahr.

From the cooling system the air was throttled into the inlet surge tank at a pressure corresponding to the air temperature for a standard altitude. The exhaust pressure was also lowered, and an altitude test was made.

Although the temperature of the air leaving the second-stage coolers could be kept at —30 deg. fahr., the lowest air temperature at the engine was —3 deg. fahr., owing to the absorption of heat from the room. This rise in temperature was obtained despite 2 in. of hair-felt lagging on the piping and surge tank. Computations indicated that additional lagging would have done very little good. With a reduction in inlet pressure and the consequent reduction in rate of flow, the lowest air temperature that could be held was 8 deg. fahr. which, in a standard atmosphere, corresponds to an altitude of 14,300 ft.[6]

The equipment just described permitted the making of altitude and auxiliary tests as discussed later. In general, the tests included simulated altitude runs from sea level to 14,000 ft., as well as runs for which the inlet-air temperature and pressure and the exhaust back pressure were controlled as single variables.

Engine Performance at Altitude

The test results presented are curves of engine performance as influenced simultaneously by the temperature and pressure conditions prevailing at altitude. Altitude temperatures were obtained by passing the inlet air through the air-cooling equipment; corresponding altitude pressures were produced by throttling the inlet air. Fig. 5(a) and Fig. 5(b) show the variation of indicated and brake engine performance, respectively, for the obtainable range of standard altitudes from sea level to 14,000 ft. Although the maximum cylinder pressure was lower for each increase in altitude, each cylinder pressure was sufficient to give maximum mean effective pressure for the subject altitude. It is noteworthy that the mean-effective-pressure curves converge to a nearly common path rather than drop as a whole for all fuel quantities. This fact permits maximum engine power to be obtained at each altitude for but a slight increase in fuel consumption.

Fig. 6 shows a comparison of the maximum engine power with altitude of a 12-cylinder unsupercharged carburetor engine and the single-cylinder data of Fig. 5(a) and Fig. 5(b). The carburetor-engine data are given on a brake basis and are believed to be representative of the best carburetor-engine performance corrected to standard altitude conditions.[7] Both brake and indicated single-cylinder performances are shown because the sea-level mechanical efficiency was only 76 per cent whereas that of the carburetor engine was 88 per cent. A multicylinder compression-ignition engine would have a sufficiently high mechanical efficiency[8] to give performance equal to or better than that of the carburetor engine.

Previous reports of the altitude performance of compression-ignition engines in aircraft have been that they lose power much less rapidly with increasing altitude than do carburetor engines. It is possible that this conclusion may have been indicated because the engines were operated with excess air at sea level so as to give a clear exhaust whereas, with increasing altitude, the air-fuel ratio was decreased toward maximum power (Fig. 5) but at the expense of a smoky exhaust. Comparison under these conditions is unfair to the carburetor engine. The results of Fig. 6 show that, for the same inlet-air conditions, the performances of the two types of engines do not differ by large amounts.

A further consideration, when comparing the altitude performance of the two types of engines, is the temperature of the inlet air. The compression-ignition engine can induct its air at the low temperatures existing at altitude and thereby obtain a maximum weight of air charge. A carburetor engine will

[6] See N.A.C.A. Technical Report No. 218, 1925; "Standard Atmosphere – Tables and Data," by Walter S. Diehl.

[7] See N.A.C.A. Technical Note No. 579, September, 1936; "Charts for Calculating the Performance of Airplanes Having Constant-Speed Propellers," by Roland J. White and Victor J. Martin.

[8] See N.A.C.A. Technical Note No. 577, August, 1936; "Friction of Compression-Ignition Engines," by Charles S. Moore and John H. Collins, Jr.

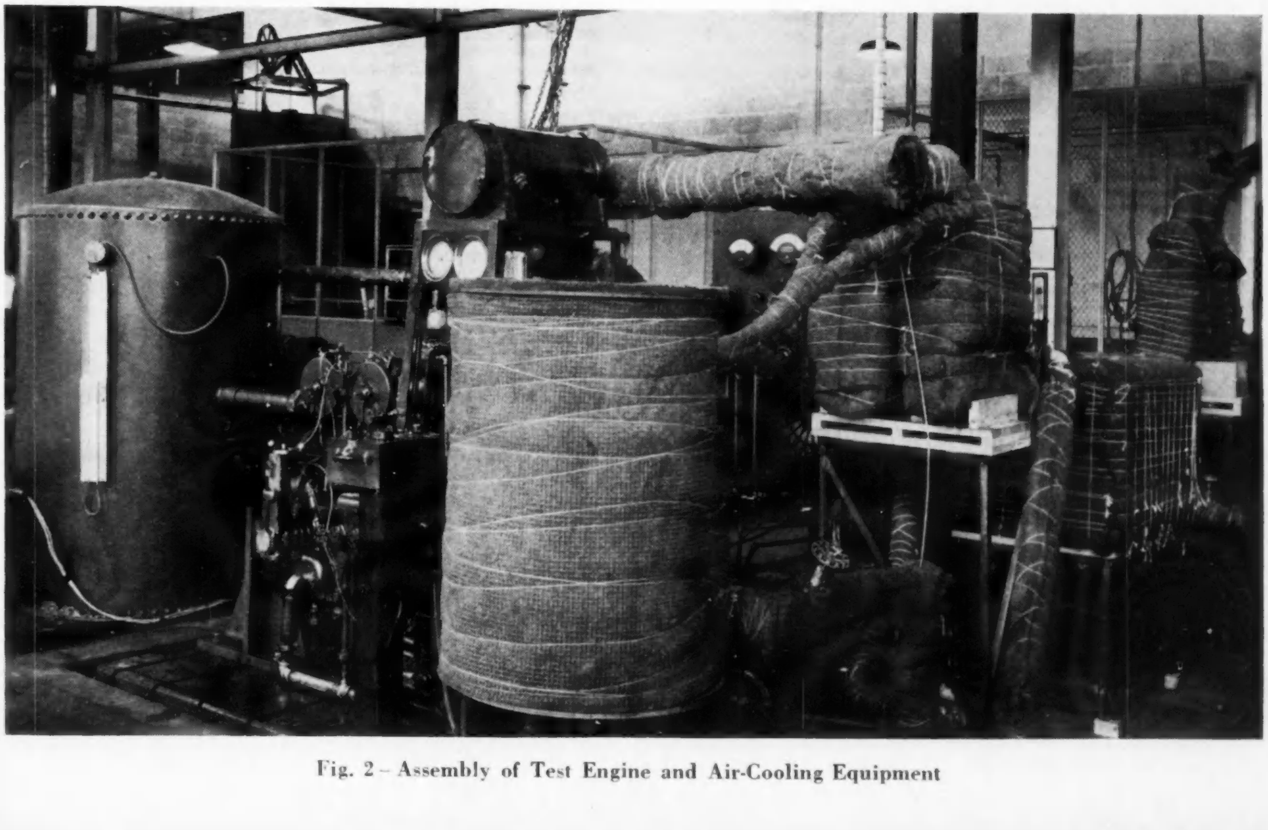


Fig. 2 – Assembly of Test Engine and Air-Cooling Equipment

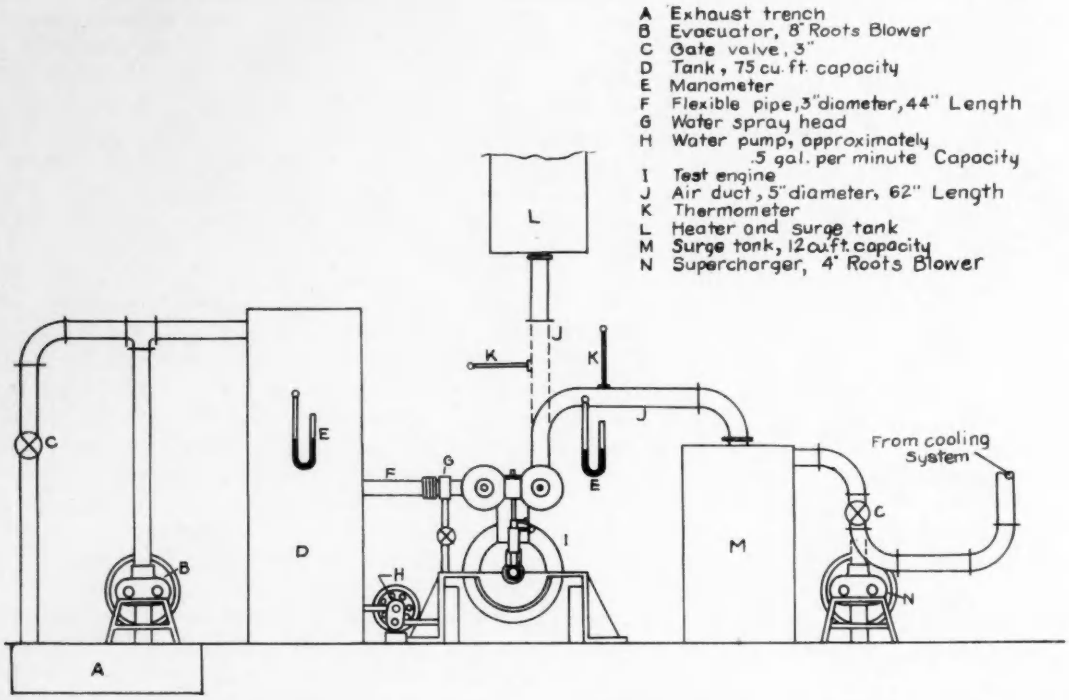


Fig. 3 – Diagrammatic Sketch of Engine and Test Equipment

have difficulty from carburetor icing if air is inducted at the low temperature of altitude. In order to prevent icing in conventional carburetors, the intake air must be heated. An alcohol spray must be used or the gasoline must be injected. If the carburetor air is heated, the weight of air charge is decreased and engine power is decreased accordingly. The curve of multicylinder carburetor-engine performance for a carburetor-air temperature of 90 deg. fahr. – the recommended temperature to prevent icing in the venturis[9] – is seen to be lower than that of the single-cylinder compression-ignition engine by about 3 per cent. For a multicylinder engine the difference in favor of the compression-ignition engine would be larger.

Induction of low-temperature air is also applicable in the case of a supercharged engine limited only by the ability of the intercooler to cool the air. The supercharged-engine curves of Fig. 6 show the performance of a two-row radial carburetor engine supercharged to 12,000 ft. compared with a compression-ignition engine having the same manifold pressure and cooling its inlet air to the standard-altitude temperatures (16 deg. fahr. at 12,000 ft.). Data for the compression-ignition curve are obtained from boosted performance and corrected to

[9] See Bureau of Aeronautics, Navy Department, Technical Order No. 37-45, Aer-E-45-MN F21-1 (23), 24 September, 1935; "Use of Carburetor Air Preheaters."

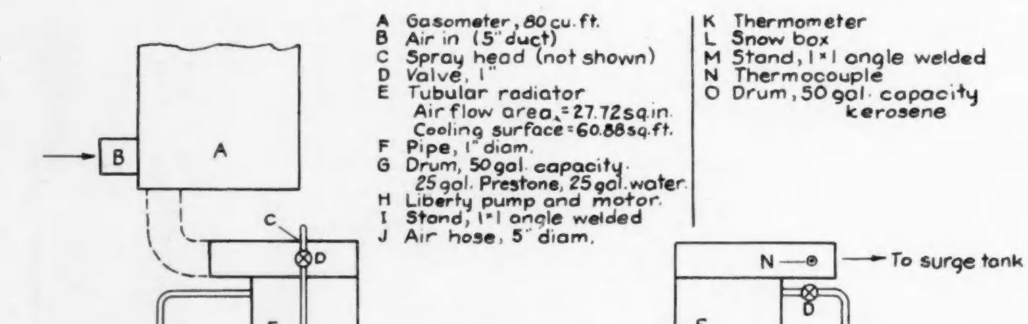


Fig. 4 – Diagrammatic Sketch of Air-Cooling Equipment

standard altitude according to test results, which will be presented and discussed later on. It is seen that, by cooling the intake air, the supercharged compression-ignition engine performance shows a possible improvement of 8 per cent at 12,000 ft.

There is a further consideration at altitudes below the critical in that the carburetor engine must limit its manifold pressure, except for short periods, in order to avoid detonation and resulting damage to the engine. On the other hand, there is no such limit to the boosting of a compression-ignition engine. Therefore the power curve may be extended downward to the right at altitudes below 12,000 ft. to give outputs in excess of the maximum shown in the figure.

Fig. 7 shows the effect of boosting at sea-level exhaust conditions. Results for similar boosts also were obtained for a series of exhaust back pressures equivalent to altitudes up to 19,000 ft. Power increases quite steadily with boost pressures but, for the pressure-rise type of combustion upon which this engine operates, it is necessary to increase the maximum cylinder pressure in order to obtain a full return from the higher boost pressure. Engine operation is very smooth for boosted operation but power is not quite optimum as maximum cylinder pressure is the limiting factor. The fuel consumption decreases at practically all fuel quantities with increase of boost pressure. The boosted tests at the altitude exhaust pressures had the same general characteristics as shown in Fig. 7, except that each decrease in exhaust back pressure caused an increase in mean effective pressure and a decrease in fuel consumption. The boosted results of this or the following curves are not corrected for power to drive the supercharger or to a multicylinder basis, factors that operate against each other.

Further boosted-altitude data are shown in Fig. 8 for which the boosted brake mean effective pressure is corrected to show the value for induction air at the standard-altitude temperature, as will be discussed later. The unboosted brake mean effective pressure decreases rapidly whereas, by boosting to constant manifold pressures and cooling the inlet air to standard altitude temperature, the brake mean effective pressure will increase with altitude. The brake mean effective pressure increases, in part, owing to diminishing exhaust back pressure but mostly owing to the possibility of inducting air at the low temperatures of altitude. The results at 35 and 45 in. of hg. manifold pressure are limiting conditions since cooling to the altitude temperature could not be achieved entirely. Furthermore, the boost is extremely high, over 30 in. of hg. above the altitude pressure at 19,000 ft., so that the cost of obtaining the boost pressure would be large. However, the gross brake mean effective pressure of about 225 lb. per sq. in. is obtainable at 19,000-ft. altitude at a maximum cylinder pressure of about 1000 lb. per sq. in.

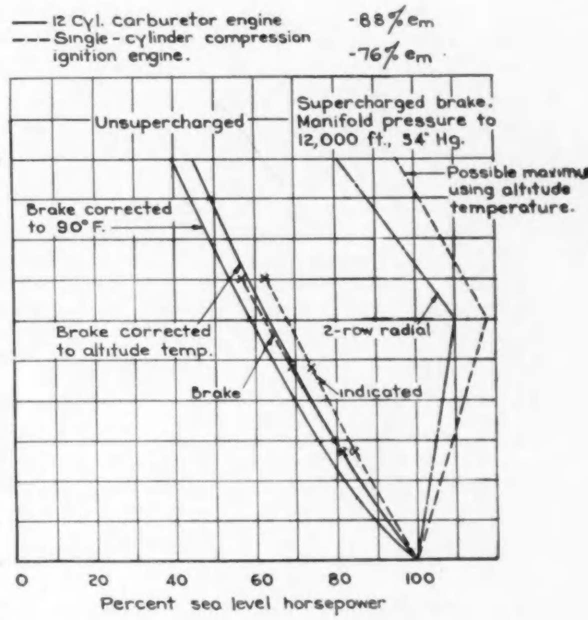


Fig. 6 – Comparison of Maximum Powers of Compression-Ignition and Carburetor Engines at Altitude

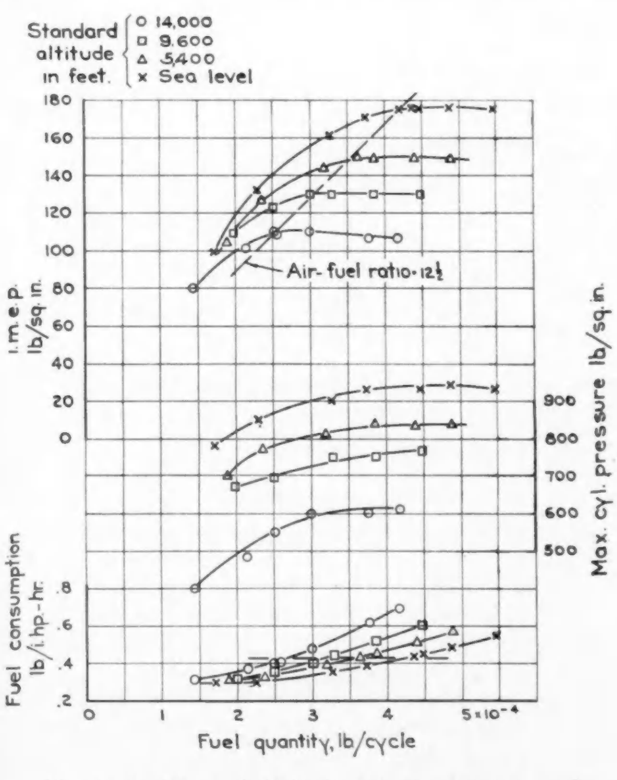


Fig. 5(a) – Effect of Altitude (Air Temperature and Pressure) on Indicated Performance

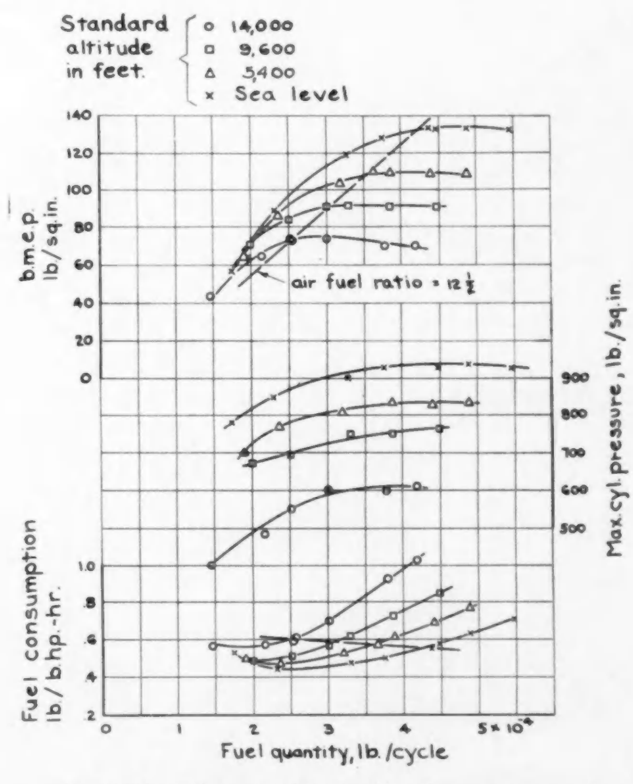


Fig. 5(b) – Effect of Altitude (Air Temperature and Pressure) on Brake Performance

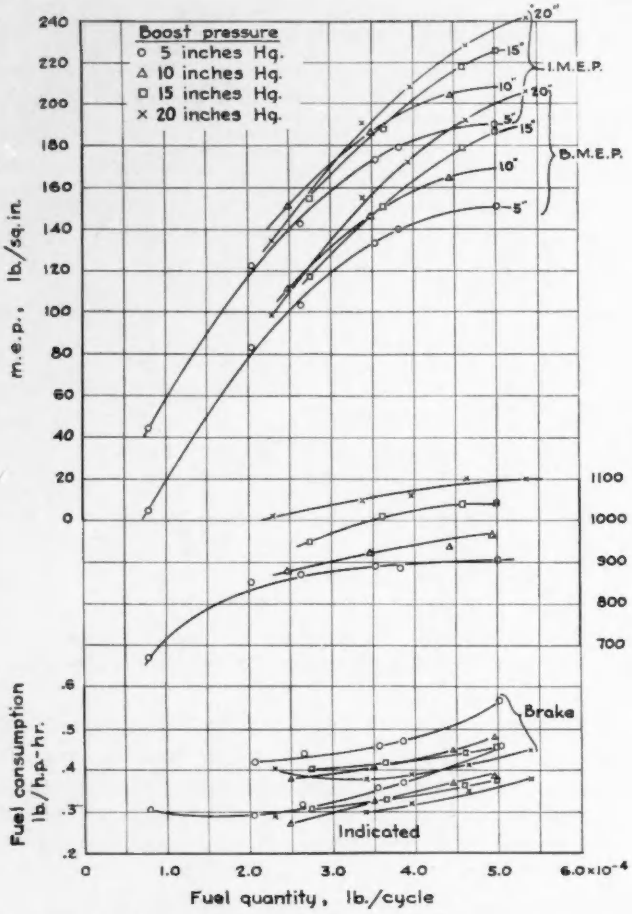


Fig. 7 – Effect of Boost Pressure; Inlet-Air Temperature, 87 Deg. Fahr.; Sea-Level Exhaust Pressure

For the simulated altitude test of Fig. 5 there was no sign of misfiring at the 14,000-ft. altitude conditions but, with the continued decrease in compression pressure, a critical altitude might be reached where compression ignition would cease. Then, either the inlet air would have to be heated or, better by far, the inlet-air pressure would have to be boosted so that more heat – but not a higher temperature – would be available to cause ignition. If the lower curve of Fig. 8 is extended to zero b. m. e. p., the critical altitude is indicated to be about 35,000 ft. for the unsupercharged condition. Probably the power would decrease more rapidly as the critical altitude is approached so that a lower altitude would be more nearly correct. In May, 1934, a Bristol "Phoenix" compression-ignition engine was flown to 27,450 ft. in an air temperature of −40 deg. fahr.[10] This engine was of 14:1 compression ratio and supercharged to 7000 ft. No sudden failure of combustion was indicated at the maximum altitude attained.

[10] See *The Oil Engine*, Vol. 2, No. 15, July, 1934, p. 77: "A British Aircraft Oil Engine's Performance."

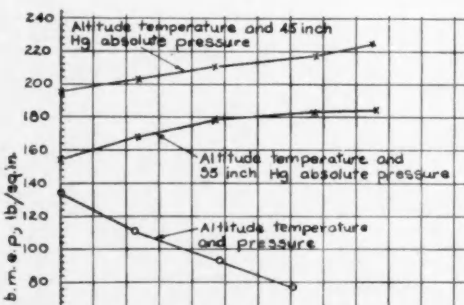


Fig. 8 – Effect of Inlet-Air Conditions at Altitude on Maximum Power; Gross, Single-Cylinder Engine

A summary of effects of altitude temperature and pressure on unboosted engine performance is shown in Fig. 9. The decreasing weight of air charge causes the motoring compression pressure to decrease and likewise causes the friction mean effective pressure to decrease. However, the mechanical efficiency also decreases because the brake mean effective pressure evidently decreases much faster than the friction mean effective pressure. The low compression pressure of 290 lb. per sq. in. at 14,000 ft. is noteworthy inasmuch as the engine operation became rough and the rates of pressure rise became extremely high.

Effect of Inlet-Air Pressure

In order to determine how engine performance is influenced by reduction in inlet-air pressure alone, variable fuel-quantity tests were made for several degrees of throttling of the inlet air to pressures as noted in Fig. 10. The temperature of the inlet air and the exhaust back pressure were kept constant at 66 deg. fahr. and 30.5 in. of hg., respectively. Engine operation was

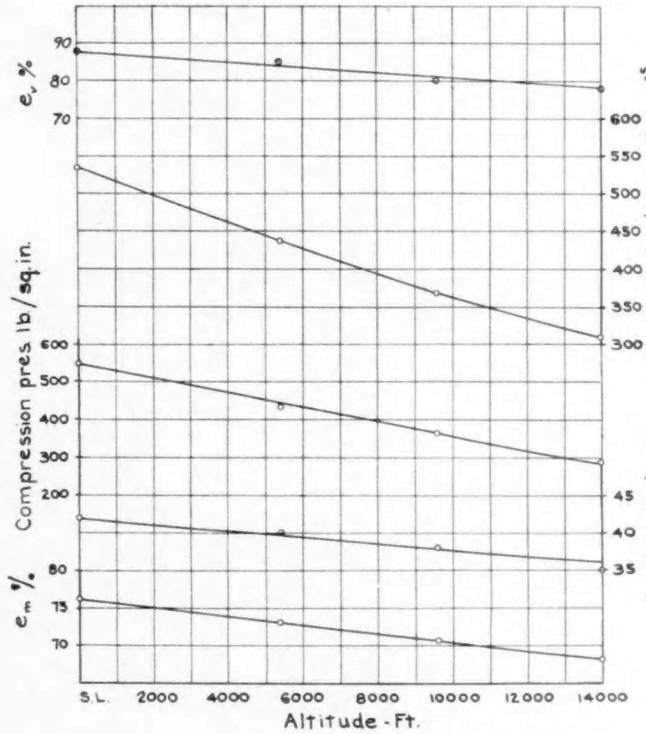


Fig. 9 – Effects of Altitude Air Temperature and Pressure

becoming rough at an equivalent altitude inlet pressure of 9,500 ft. and a run at 15,000 ft. inlet pressure could not be made because the engine would not maintain the test speed of 2000 r.p.m. even without load. For the unnatural conditions of this test, then, the imaginary critical altitude is between 9,500 and 15,000 ft. whereas, for the altitude temperature and pressure tests, the increased air charge obtained at the low inlet-air temperature and permitted by the reduced exhaust back pressure raised the critical altitude to well over 14,000 ft. The variable fuel-quantity curves have the same characteristics as those of the altitude tests and show the same convergence at decreasing fuel quantity and inlet-air pressure. Although the maximum cylinder pressures are much different for the different inlet pressures, the mean effective pressures are optimum in all cases. The mean effective pressures plotted in Fig. 11 are the maximum obtainable, and the points at pressures less

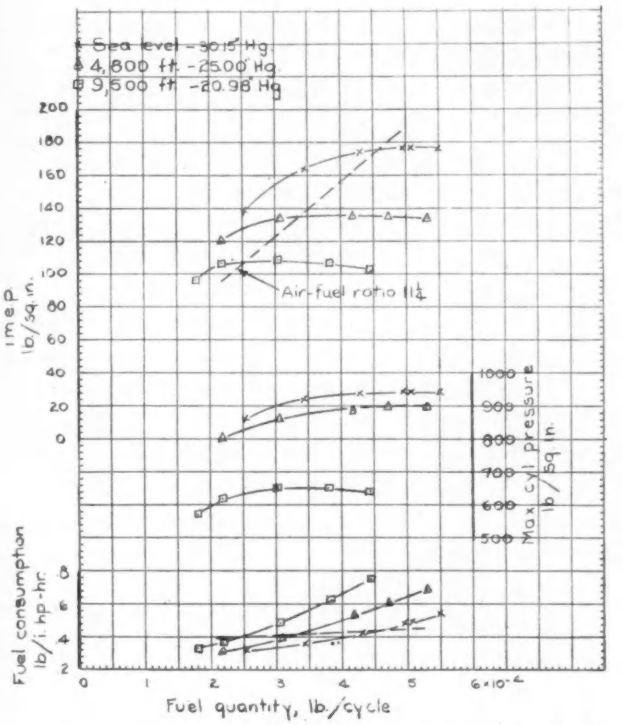


Fig. 10 – Effect of Reduced Inlet-Air Pressure; Inlet-Air Temperature 66 Deg. Fahr., Sea-Level Exhaust-Back Pressure

than sea level were taken from Fig. 10 at the lowest fuel quantity giving the maximum power. It so happened that the air-fuel ratio was nearly constant for these maximum points. The booster section of the curves of Fig. 11 is not for optimum power but is limited by maximum cylinder pressure and was obtained from Fig. 7. The slope of the boosted curves can be

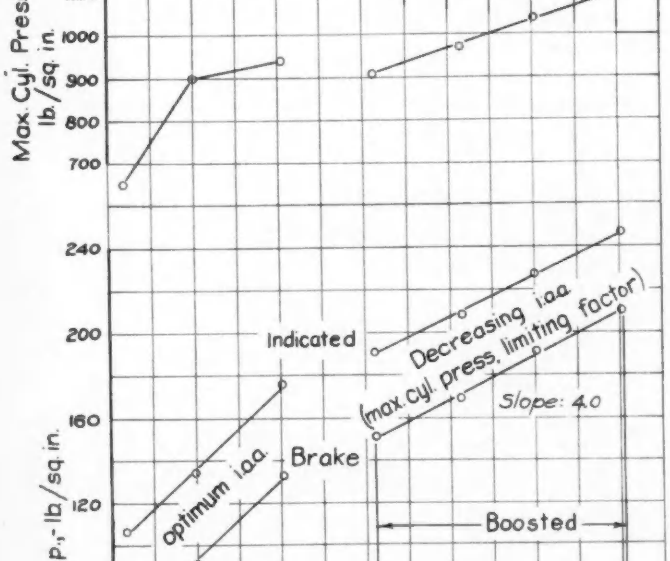


Fig. 11 – Effect of Inlet-Air Pressure on Maximum Sea-Level Performance – Temperature 66 Deg. Fahr.

influenced by maximum cylinder pressure but the slope of the throttled pressure curves is believed to be influenced only by inlet-air pressure. Although only three points were taken for the test presented, additional results also indicate that the trend with inlet-air pressure is practically a straight line of 7.4 lb. per sq. in. increase in mean effective pressure for 1 in. of hg. increase in inlet-air pressure.

Effect of Inlet-Air Temperature

The variable fuel-quantity results of Fig. 12 were obtained by using the air-cooling and air-heating equipment. During these tests the barometric pressure variation was ± 0.4 per cent while the experimental error was approximately ± 1 per cent. The test results show, then, the effect of inlet-air temperature alone for a practically constant air pressure. The mean-effective-pressure and fuel-consumption curves have the same convergence as the altitude and inlet-air-pressure-variation curves previously discussed. Maximum power for each temperature run of Fig. 12 occurs at an air-fuel ratio that is nearly constant at $12\frac{1}{2}$. When plotted against temperature in Fig. 13, the maximum mean effective pressure is seen to vary nearly as a straight line except for the point at lowest temperature. An analysis of the test results indicated that this point was too low in performance, and the curve accordingly was drawn high. For purposes of correcting engine performance for differences in inlet-air temperature, the indicated and brake mean effective pressure variation of Fig. 13 may be considered as straight lines and the slopes used to correct the mean effective pressure. For the engine tested the slopes were 0.226 lb. per sq. in.

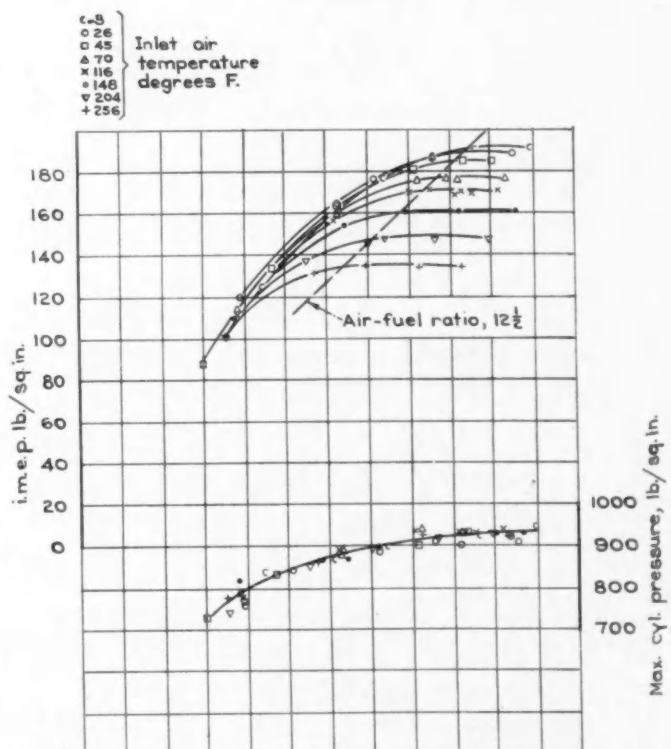


Fig. 12 – Effect of Inlet-Air Temperature on Sea-Level Performance

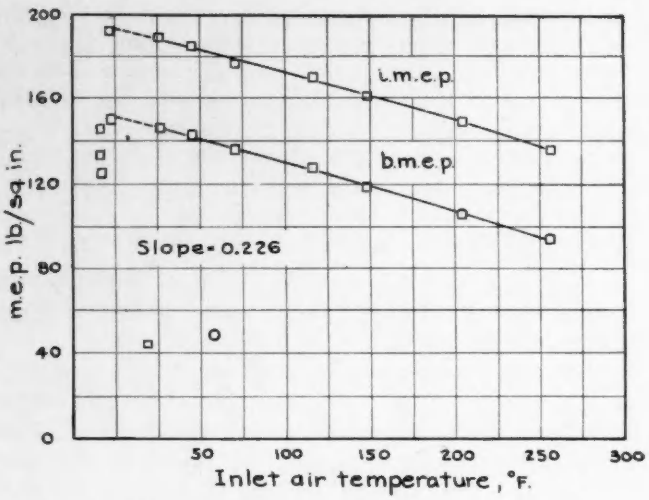


Fig. 13 – Effect of Inlet-Air Temperature on Maximum Performance at Sea Level

increase in m.e.p. for each deg. fahr. decrease for both the indicated and brake performance.

A summary of subordinate test results similar to Fig. 9 is shown in Fig. 14 for both pressure and temperature variations. In the analysis of these curves it should be remembered that increasing air temperature and air pressure have opposite effects on air density and weight of air charge. This fact explains why the curves of Fig. 14 are of opposite slopes and, in most cases, cross each other. Although the injection advance angle was increased with increasing inlet-air temperature to maintain a constant maximum cylinder pressure, the engine operation did not become rough. In fact, engine operation was smooth throughout the range of temperatures investigated. A section of the air weight-pressure curve is shown as a broken line because, at air pressure greater than standard sea level, the overlap of the engine valves permitted some air to pass through the engine so that the actual air-charge weight was less than the measured weight. A volumetric efficiency of 100 per cent

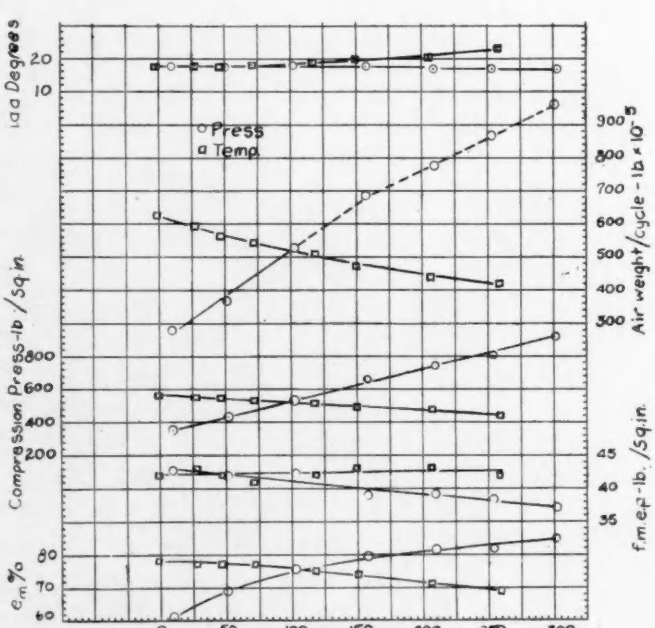


Fig. 14 – Effects of Inlet-Air Pressure and Temperature at Sea Level

was assumed for such inlet conditions, and weights of boosted air charge were calculated on this basis. Variation of compression pressure, friction mean effective pressure, and mechanical efficiency follow in most cases from the variation of weight of air charge.

In most engine-performance corrections, air-fuel ratio and inlet-air density are considered fundamental factors. In this paper only maximum engine performance has been considered, therefore any air-fuel ratio would not have given more power. It was thought, however, that engine performance might vary directly with inlet-air density, and Fig. 15 was plotted of maximum engine performance for inlet-air densities obtained by independently varying the temperature and pressure. For the same air densities the mean-effective-pressure curves are displaced one from another and are of different slopes depending upon how the inlet-air density was obtained. Evidently density of inlet air is not the factor controlling maximum

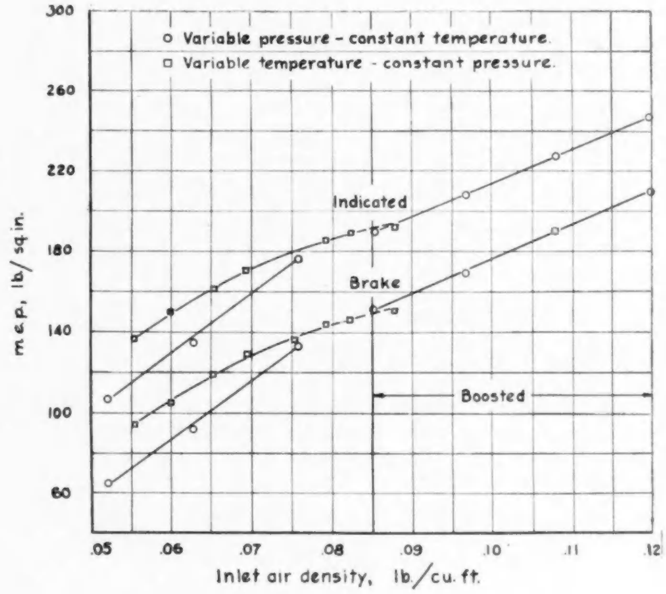


Fig. 15 – Effect of Inlet-Air Density on Maximum Mean Effective Pressure at Sea Level

engine power. Continuing the analysis further, it was thought that perhaps the volumetric efficiency and, consequently, the air-charge weight, was different in the three cases. Fig. 16 was prepared on a weight of air charge basis and better agreement is shown over a longer section of the curves. At low weights of air charge as given by heating the inlet air there still is increasing divergence of the curves. Apparently weight of air charge is a better method of correction than density of inlet air, but even then there is not complete agreement for corrections at high inlet-air temperatures.

The variations of power for the boosted section of the curves are straight lines because the maximum cylinder pressure was increased at a constant rate with boost pressure and was a limiting factor for mean effective pressure.

In this paper only maximum engine performance has been considered regardless of air-fuel ratio. The air-fuel ratio for the altitude test maximum performance was 12½, for the inlet-pressure variation 11¼, and for the inlet-air temperature variation, 12½. If a constant air-fuel ratio had been chosen the performances obtained would not have been maximums in all cases. Neither would performances at the same air-fuel ratio have been equal on the air-density or weight-of-air-charge curves. For example, in Fig. 16 if a constant air-fuel ratio had been taken, the curves would be separated rather than in

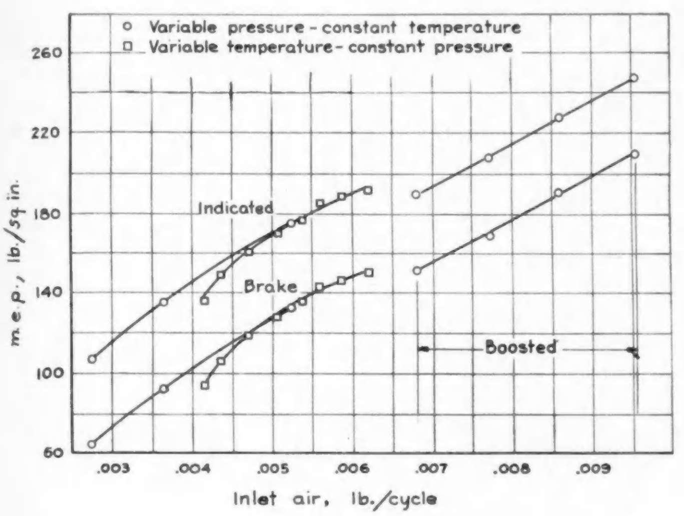


Fig. 16 – Effect of Weight of Air Charge on Maximum Mean Effective Pressure at Sea Level

partial agreement. However, an analysis on an air-fuel basis undoubtedly would yield some interesting results, especially at excess-air air-fuel ratios.

Variation in humidity is not believed to have influenced the test results because the maximum variation in weight of water vapor was only ±0.6 per cent. Carburetor-engine tests[11] have indicated that humidity affects engine power only to the extent of displacing air with water vapor and thus reducing the weight of air available for combustion. In the results presented by Brooks and Garlock[11] the variation of humidity was less than the experimental error.

Effect of Exhaust Back Pressure

The effect of exhaust back pressure on the performance of a compression-ignition engine is important; first, because back pressure decreases with altitude and, second, back pressure increases with the application of an exhaust-driven turbo-centrifugal supercharger. Fig. 17 shows the effect on the mean effective pressures, varying the exhaust back pressure as a single variable. The curves are discontinuous at sea-level pressure because the curves are from two series of tests made on different days and at different maximum cylinder pressures, as noted on the figure. With decreasing back pressure the steady reduction in friction mean effective pressure causes most of the improvement in brake mean effective pressure. The indicated curve is variable in trend, probably because the decreasing back pressure first helps to improve clearance-volume scavenging and to increase the weight of air charge and then, finally, the large pressure difference during the valve-overlap period upsets the ensuing air charging. For pressures greater than prevailing at sea level the engine performance is affected adversely and at an increasing rate. But, if the exhaust pressure were used to drive a turbo-supercharger at altitude, the mean-effective-pressure loss to exhaust back pressure would be somewhat reduced by the decreased atmospheric pressure.

Indicator Cards

Fig. 18 shows five indicator cards obtained while the engine was operating at the several conditions of this series of tests. When studying these indicator cards, it must be remembered that widely different inlet-air temperatures and pressures were used. Both weight of air charge and weight of fuel charge were varied with each condition so that an indicated-mean-effective-pressure comparison is usually not possible. The center card (*c*) was taken while the engine was operating at normal unboosted sea-level conditions and is presented for purposes of comparison. For card (*c*) engine operation was smooth and regular. Comparing card (*a*) with card (*c*) shows the effect of altitude on the indicator card. The compression pressure is lowered, although it is about 50 lb. per sq. in. higher than for the motoring compression data of Fig. 9 owing to heat absorption from the cylinder walls. This lowered compression pressure caused the ignition lag to increase, as can be seen from the cards, and permitted a greater accumulation of fuel prior to ignition. Then on ignition the rate of pressure rise is higher and is nearly a straight line as the rough operation previously noted had indicated. The high maximum cylinder pressures are of little use because they do not increase the area of the indicator card.

Card (*b*) was taken while the engine was operating with sea-level inlet air and exhaust back pressure but with an inlet-air temperature of 1 deg. fahr. Ignition lag is affected little, if any, by the low inlet-air temperature while smoothness of operation was improved further, as the lowered rate of pressure rise would indicate.

Effects of high inlet-air temperature are shown by cards (*d*) and (*e*), the difference between the two cards being maximum cylinder pressure and rate of pressure rise as controlled by injection advance angle. Card (*d*) is for the same injection advance angle as card (*c*) and shows the effect of an inlet-air temperature change from 258 deg. fahr. to 71 deg. fahr. At the higher temperature the ignition lag was decreased which, in turn, decreased the fuel accumulated at ignition and decreased the ensuing rate of pressure rise. Even the 23½ deg. injection advance angle of card (*e*) did not give rough operation even with the early pressure rise that the card shows. The high inlet-air temperature and correspondingly short ignition lag prevented excessive fuel accumulation so that the rate of pressure rise is relatively low.

Conclusions

The series of tests just described indicate:

(1) The altitude performance of both a supercharged and an unsupercharged compression-ignition engine would compare advantageously with a carburetor engine; the low temperatures of altitude are especially important in maintaining the altitude power of the compression-ignition engine.

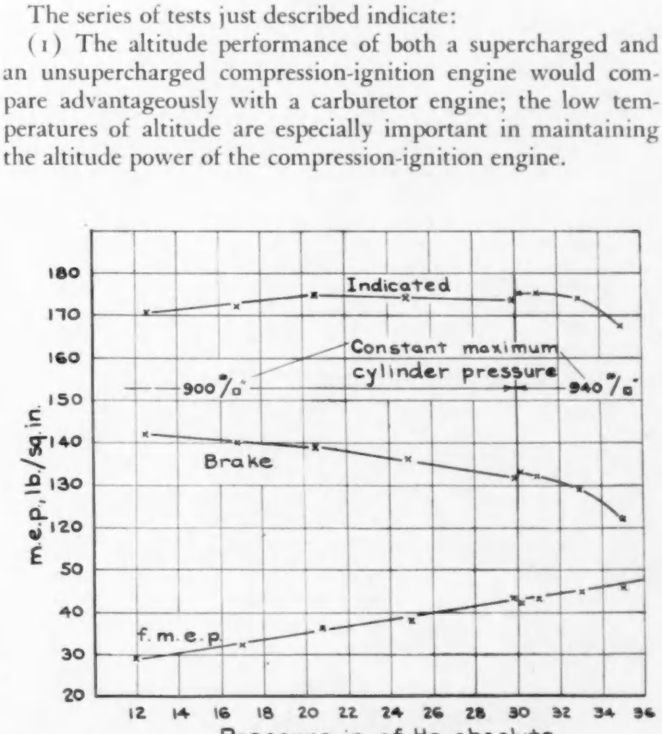
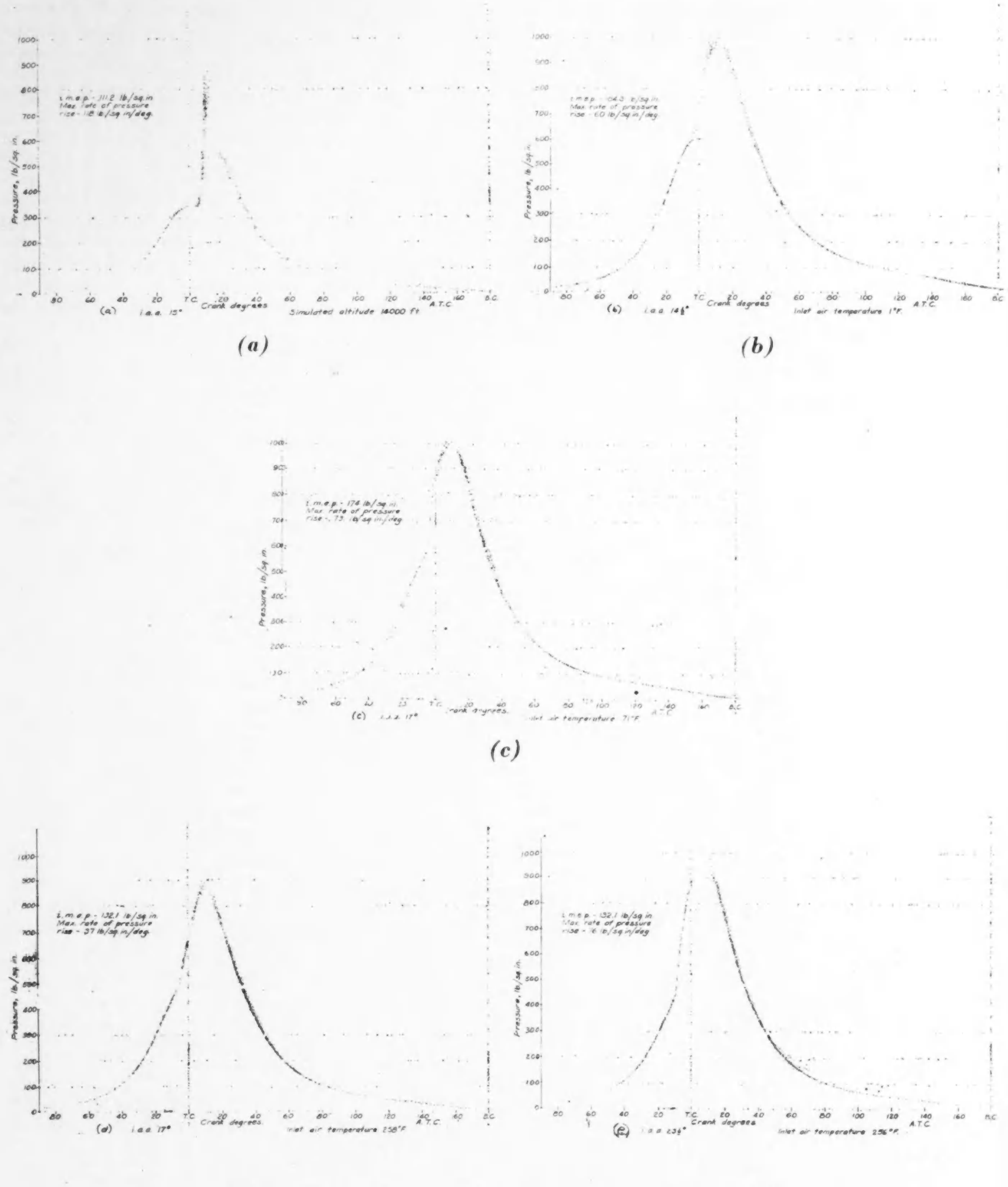


Fig. 17 – Effect of Exhaust Back Pressure on Maximum Mean Effective Pressure – Sea-Level Inlet Pressure

[11] See N.A.C.A. Technical Report No. 426, 1932; "The Effect of Humidity on Engine Power at Altitude," by D. B. Brooks and E. A. Garlock.



(*d*) **Fig. 18 – Indicator Cards for Various Test Conditions** (*e*)

(2) Maximum performance of this unsupercharged compression-ignition engine cannot be corrected accurately on an inlet-air-density or weight-of-air-charge basis for differences of inlet-air temperature and pressure.

(3) Maximum unsupercharged performance can be corrected, when maximum cylinder pressure does not limit output, as follows:

For each inch of hg. increase in inlet-air pressure add 7.4 lb. per sq. in. indicated or brake mean effective pressure.

For each deg. fahr. increase in inlet-air temperature subtract 0.226 lb. per sq. in. indicated or brake mean effective pressure.

(4) Maximum boosted performance with conservative maximum cylinder pressures can be corrected as follows:

For each inch of hg. increase in inlet-air pressure add 4.0 lb. per sq. in. indicated or brake mean effective pressure.

(5) Reduced exhaust back pressure increased engine power slightly, whereas increased exhaust back pressure decreased engine power at an increasing rate.